A PRACTICAL TREATISE ON THE THEORY, DESIGN, CONSTRUCTION AND USE OF THE MODERN STEAM ENGINE.

THE MODERN STEAM ENGINE

THEORY—DESIGN—CONSTRUCTION—USE

A Practical Treatise

BY

JOHN RICHARDSON

M.Inst.C.E., M.Inst.M.E., Consulting Engineer

With Three Hundred Illustrations

NEW YORK

D. VAN NOSTRAND COMPANY

23 MURRAY AND 27 WARREN STREETS

1908

BUTLER & TANNER,
THE SELWOOD PRINTING WORKS,
FROME, AND LONDON.

Preface

As this book is intended principally for use of those who have no previous knowledge of the subject, I have begun at the beginning and have carefully confined myself to the use of terms, formulas, and illustrations, of the simplest character, and such as can be easily understood by all.

For those students who wish to tackle very abstruse problems, needing the use of the higher mathematics, there are many suitable books already published, and to these such students are referred.

Such a course of study is for the very few; the contents of this book are for the much larger class, who, starting without knowledge, want to obtain a clear but comprehensive acquaintance with the steam engine.

In writing the book I have drawn principally upon a long practical experience extending over more than forty years, during which time I have superintended the designs and construction of over 25,000 steam engines of many kinds; and varying from 5 to 2,000 horse-power each.

I wish nevertheless to render my very sincere thanks to those eminent engineers who, for the purpose of this book, have prepared and sent me drawings and particulars of their special manufactures, without which this work would have been very incomplete.

In some technical schools and colleges I have not infrequently seen drawings of work for students to copy, which have been prepared by high theoretical authorities, but which showed not only faulty, but sometimes impossible methods of manufacture. It is only by a long practical experience that

such teaching can be avoided. Further, it must be remembered that though certain things may be possible and good, yet there may be other and better ways of designing machinery and doing work. It is these better ways I wish to teach.

The method of instruction I have adopted is that which I have found most effective in conveying ideas and information to the many hundreds of youths I have had under my engineering training during the last forty years; my object being not to cram them with dry facts and formulas, but to make them thoroughly understand the why and wherefore of all tables and formulas used, and how to be independent of them at need.

Users of steam engines as well as students will find valuable help and information in the following pages, the sum of all being to teach all who design, construct, or use steam engines, what to avoid, what to do, and how to do it.

JOHN RICHARDSON.

LINCOLN,
October 23, 1907.

Contents

CHAPTER I

UNIT OF POWER

CHAPTER II

NATURAL FORCES

CHAPTER III

EARLY TYPES OF ENGINE

CHAPTER IV

THE WATT ENGINE

CHAPTER V

THE USE OF STEAM IN MULTIPLE CYLINDER OR COMPOUND ENGINES

CHAPTER VI

VALVE GEARS

CHAPTER VII

DOUBLE OR COMPOUND SLIDE VALVES

PAGES

CHAPTER VIII

RELEASING VALVE GEAR

CHAPTER IX

EXHAUST VALVES AND VALVE DRIVING GEAR

CHAPTER XIV

DESIGN OF DETAILS

CHAPTER XV

EXAMPLES OF VARIOUS TYPES

CHAPTER XVI

FEED AND STEAM HEATING

CHAPTER XVII

HINTS TO USERS

APPENDIX

CHAPTER I

Unit of Power

AMONGST the first things we need to understand are :—What is a horse-power ? What is a foot-pound, and why are 33,000 foot-pounds called one horse-power ?

What is a Horse-Power ?

It may be stated at once that the unit of measurement, like many others we have to use, is an inconvenient and unsatisfactory one, but its use is so thoroughly established in all parts of the civilized world, that it will be many years before it will be replaced by a better one, and in the meantime we must not only know, but also thoroughly understand it.

Power of Horses.

Many years ago, before steam engines came into use, horses were very largely used to do work in driving machinery and lifting materials as well as in drawing carts and carriages along the roads as they do now. A simple and well-known device by which a horse-power was utilized in lifting weights is shown in the sketch below (Fig. 1).

FIG. 1.

Primitive Method of Raising Weights.

Here it will be seen a sack of corn is being hoisted to the top story of a high warehouse by means of a rope going over a pulley, one end of the rope being attached to the sack and the other end to the traces of a horse running on the ground. The man with

the horse keeps his eye on the sack, and when it gets through the trap-door, the horse is stopped, the sack released and the rope lowered again for the next lift, the horse trotting back in the meantime. The horse soon got used to its work, and would start at a signal from the man, stop when the sack got to the top, back a little for its return on to the platform, and trot back again for its next load without any man to guide it. Now a sack of corn weighs about 20 stone, and 20 × 14 = 280 lb. This weight, plus the weight of an iron ball, hung on the rope above the hook, which was required to lower the rope and pull back the sack, altogether making up a weight of say 330 lb. Supposing the warehouse to be 50 feet high, it was found that 2 sacks a minute could be hoisted up by this means; and, if it was only 25 feet high from the wagon to the top floor, then 4 sacks a minute could be hoisted up.

Problems in Horse-Power.

Thus :—

330 lb. × 25 feet × 4 times a minute = 33,000
330 „ × 50 „ × 2 „ „ = 33,000
330 „ × 100 „ × 1 „ „ = 33,000

It will be noticed that all these sums give the same result, 33,000, they are the result of multiplying pounds by the number of feet raised in a minute and are called *foot-pounds*; this is one unit of measurement and 33,000 of them equal one horse-power.

In the illustration we have given, we have taken the same weight, 330 lb. lifted to different heights, but all at the rate or speed of 100 feet per minute, but it must be obvious that if the weight were much less, the horse instead of walking slowly could go more quickly; thus, if the load were only one-half of 330 lb. there would be only one-half the strain upon it and it could go twice as fast. Let the sack then be replaced by a bale of goods weighing only 165 lb. Then twice as many bales could be lifted in the same time. The horse, instead of pulling hard at the rate of say 2 miles an hour, would go along nimbly at 4 miles an hour. Now there are 1,760 yards in a mile or 5,280 feet. 5,280 feet × 4 (miles an hour) equals 21,120 feet, which divided by 60 (minutes in an hour) gives us 352 feet per minute, or nearly 100 feet per minute for every mile per hour (exactly 88). (This is a fact which may be remembered and which will be found useful in

Effect of Speed on Power.

making rapid mental calculations for obtaining approximate results.)

The horse then with its light load went at a little over 4 miles an hour or 400 feet per minute, but though it trotted back more quickly still on its return journey, yet it did no useful work, only half its time being occupied in lifting. It would lift, theretore, through 400 feet in 2 minutes or at the average rate of 200 feet in one minute.

Now, taking the same form of illustration with this smaller load we find that—

165 (lb.)	× 25 (feet high)	× 8 (times in a minute)			= 33,000
165 ,,	× 50 ,, ,,	× 4 ,,	,,	,,	= 33,000
165 ,,	× 100 ,, ,,	× 2 ,,	,,	,,	= 33,000
165 ,,	× 200 ,, ,,	× 1 ,,	,,	,,	= 33,000

the same as before.

Again, suppose we have a much heavier load instead of 330 lb., let it be 660 lb. In this case one horse could not regularly lift it. We should require 2 horses pulling together and going say at 200 feet in a minute, which, allowing for the return journey, gives a useful 100 feet per minute.

660 (lb.)	× 25 (feet high)	× 4 (times in a minute)			= 66,000
660 ,,	× 50 ,, ,,	× 2 ,,	,,	,,	= 66,000
660 ,,	× 10 ,, ,,	× 1 ,,	,,	,,	= 66,000

This is the work done by 2 horses and the sum of 66,000 divided by 2 = 33,000 foot-pounds, which is one horse-power.

Standard Unit of Power. It will be seen that the standard unit of power, 33,000 foot-pounds, is a compound of three different things or "factors" as they are called ("factor" from facio, to make, "a maker"). These three things are "weight," or resistance, "distance" in feet, and "time" in minutes. These two last are frequently combined in one, which then becomes "speed" in feet per minute.

These factors may be multiplied together in any way and the total sum arrived at, divided by 33,000, will always give the result in horse-power. It is essential to get a very clear idea of how a horse-power is arrived at, as it will be very frequently used in the following pages.

It will be seen on consideration that, in the first illustration, that of a sack of corn going up at the average rate of 100 feet per minute, the sack really went up at a much higher speed than this, something like 250 feet a minute, but it was not

going up all the time; after the sack was lifted, the rope had to be lowered down again, the horse had to come back to a fresh start, the rope had to be attached to a fresh sack, and assuming the height to be 50 feet the operation would be something like this: the horse would walk on at the rate of say 250 feet per minute, so 50 feet would occupy one-fifth of a minute in time, i.e. 12 seconds, the return journey and the attachment of second sack say 18 seconds = together, 30 seconds, or half a minute; so that 2 sacks would be taken in one minute to a height of 50 feet, though only 24 seconds would be occupied in lifting them.

In calculating then the total horse-power to do a certain amount of work, we must take into account the total weight lifted per minute, in a consecutive series of minutes, leaving for the moment out of consideration the exact time in which it is lifted.

Intermittent and Continuous Work.

We have dealt with a horse overcoming a pull of 330 lb. and doing intermittent work. If instead of this intermittent work, the horse was required to pull continuously, like drawing a load on a level road, or driving machinery by means of a horse gear, walking at 330 feet a minute, then it could only pull 100 (lb.) and 100 (lb.) $\times$ 330 (feet per minute) = 33,000 foot-pounds.

That the student may be sure that he quite understands the foregoing a few examples like the following should be worked.

Required the nett horse-power to raise 20 tons of grain to the upper floor of a warehouse which is 50 above the ground.

20 tons of corn at 2,240 lb. per ton = 44,800 lb. This has to be raised in an hour. Divide the amount therefore by the number of minutes in an hour, i.e. 60, and we have $\frac{44,800}{60} =$ 746·6 to lift in one minute 50 feet high; thus—

$$\frac{746{\cdot}6 \times 50}{33,000} = 1{\cdot}13 \text{ H.P.}$$

Take it another way. How many tons could one horse-power raise 50 feet high in an hour?

One horse-power = 33,000 foot-pounds, this multiplied by 60 minutes = 1,980,000 foot-pounds, i.e. 1,980,000 lb. raised 1

foot high ; as we have to deal with 50 feet, this amount must be divided by 50 ; $\frac{1,980,000}{50}$=59,600 lb. and $\frac{39,600}{2,400}$ (the number of lb. in a ton)=17·8 tons.

In the earlier illustrations we have given we have assumed a pull of 350 lb. on the horse's traces. This is a heavy pull for a horse and could not be endured were it continuous. But a horse can pull harder than this for a short time. Over an

Road Traction. ordinary road a cart horse can draw a ton load say with the cart, 25 cwts. or 2,800 lb. Now the horse does not have to pull with this force or anything like it. On a level road all that it has to do is to overcome the friction of the wheels upon the axle and upon the surface of the road. Were the road perfectly smooth and hard like a railroad is, the latter friction would be only 7 to 8 lb. per ton, but upon ordi-

Friction. nary roads such as we have, it would be for a very good asphalt road about 20 lb. per ton, on a good macadamized road 40 to 50 lb. per ton, and on a country road 70 to 80 lb. per ton up to even 100 lb. on a rough road. For average roads it is usual to take 80 lb. per ton, and thus the load being 25 cwts. or 1·25 tons, the ordinary pull on the level would be 80 lb. × 1·25 (ton) or 100 lb. pull on the horse's collar and traces. This is very easy work for a horse, but roads are seldom level and may incline upwards for long dis-

Gravity. tances at the rate of 1 foot rise for 16 feet horizontal ; this is called 1 in 16. Bridges have to be crossed, and they may be for a short distance as steep as 1 in 10 ; thus—

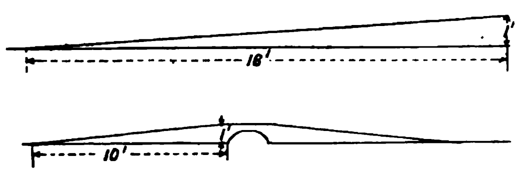


Fig. 2.

We want to find out how much harder the horse will have to pull in these two cases. In the first case 1 in 16 ; it will be seen that for every 16 feet travelled horizontally the weight is raised 1 foot high. We have already learned that to raise 1 lb. 16 feet high, or 16 lb. 1 foot high, take exactly the same power. So, if we take $\frac{1}{16}$ of the weight being lifted the whole

distance and add this amount to the pull due to friction, we shall then find the whole pull on the horse : now $\frac{2,800}{16}=175$, and this is the extra pull due to gravity ; the pull in this case has been caused by the attraction of the earth to the load, which attraction has to be overcome when we go uphill. Now, assuming that the horse is going at the rate of 250 feet a minute, about $2\frac{3}{4}$ miles an hour, we have—

Load due to friction	. . .	100 lb.
,, ,, ,, gravity	. . .	175 ,,
	Total .	275 lb. pull.

$\frac{275\times 250 \text{ (feet per minute)}}{33,000}=2{\cdot}08$, i.e. a shade over 2 horse-power. Just while we are going over the bridge, the pull will be still heavier. In this case we shall have to deal not with $\frac{1}{16}$ but with $\frac{1}{10}$ of the total weight, viz., 2,800 divided by 10= 280 plus 100 friction—

$$\frac{380\times 250 \text{ (feet per minute)}}{33,000}$$

which gives us no less than 2·87, i.e. nearly 3 horse-power, but the horse can still do this as the extra strain lasts but for a short time. The ordinary strain on the level is only $\frac{100\times 250}{33,000}=\cdot 75$, or but $\frac{3}{4}$ of a horse-power.

Inclines and Declines.

Again, for nearly every incline upwards, there is a corresponding decline downwards, when the horse can practically rest, the load going on by itself due to action of gravity. Thus going down a hill of 1 in 16 the load against the horse due to friction is for the $1\frac{1}{4}$ ton 100 lb., whereas the propelling power due to gravity is 175. Thus instead of having to pull, the horse is being really pushed forward by a force of 75 lb. through the backband. This makes comfortable travelling for the horse when the decline is not more than 1 in 16. But when the load comes down the declivity of a bridge 1 in 10, then there is a big push which has to be resisted with severe force. Here we have friction 100 lb. resisting the motion, but gravity equal to a pressure of 280 lb. aiding the motion ; thus the nett force pushing the horse downhill is 180 lb., and it takes quite as much power to resist the force as if the horse were pushing or

pulling against a force of 180 lb. uphill, at the rate of say 250 feet per minute; this would$=\frac{250\times 180}{33,000}=1{\cdot}36$ H.P. Now it is a dangerous as well as a foolish thing to be straining a horse for nothing, so if a long steep hill has to be descended, means are taken to increase the friction and therefore the resistance—the wheel may be locked by a chain, a slipper put under one or more wheels, or the loaded cart may be tilted back upon a drag-pole, these ways being merely different methods of increasing friction, and thus retarding the vehicle, as in Fig. 3.

Methods of Increasing Friction.

FIG. 3.

Riders of bicycles will all have noticed the same thing. As roads frequented by them are generally good, the friction will not be more than 50 lb. per ton, and the weight of load including cycle, about $\frac{1}{14}$ of a ton, say 150 lb., the speed of nearly 10 miles an hour or say 800 feet per minute. As we have only $\frac{1}{14}$ of a ton the road friction will be $50\div 14$, or 3·4 lb.; thus 3·4 lb. is all the push that is required to move a bicycle with its rider on a good level road. Thus to get the power we have $\frac{3{\cdot}4\times 800}{33,000}={\cdot}08$ H.P., that is about $\frac{1}{12}$ of a horse-power or, as we reckon, 6 men are equal to one horse, $\frac{1}{2}$ of a man-power. When, however, we tackle a hill, the case is very different, let the hill be 1 in 10, then we have to add to our load $\frac{1}{10}$ of the total weight of 150 lb., i.e. 15 lb., $15+3{\cdot}4=18{\cdot}4$ lb., then $\frac{18{\cdot}4\times 800}{33,000}={\cdot}45$ H.P., or practically the power of 3 men if the speed be kept up; and, even if not, yet the strains are 6 times as great. A man having, like a horse, a large margin of power for a short time, such a hill can be surmounted if it be a short one; but it will be seen from the foregoing that the cyclist has to work 6 times as hard in ascending a hill of 1 in 10 as he does on a good level road.

Bicycle Illustration.

The horse, being an animal with a large reserve of strength, can seem to do more work than a steam engine of equal power. Thus 4 horses will draw a wagon loaded with 5 tons over a

short hill say even 1 in 10 at 250 feet per minute, while it would require a traction engine of much more than this power to do the same work; thus—

5 tons plus 1 ton for the waggon	=6 tons, or 13,440 lb.
Gravity at 1 in 10 would	=1,344 lb.
Friction of 6 tons at 80 lb.	= 480 lb.
Total	1,824 lb.

$$\frac{1{,}824 \text{ (lb.)} \times 250 \text{ (feet per minute)}}{33{,}000} = 13{\cdot}8 \text{ H.P.}, \text{ this being the}$$

nett power the 4 horses exert for a short time.

Traction Engine Work.

Now if we take the same load drawn by a road traction engine the engine with its tender, coals and water will, in addition to the 6-ton load, weigh at least another 6 tons; thus we should have 12 tons in all or 26,880 lb.

Thus for load due to friction we should have—

12 tons at 80 lb.	= 960 lb.
Gravity 1 in 10	=2,688 lb.
Total	3,648 lb.

$$\text{Thus } \frac{3{,}648 \times 250}{33{,}000} = 27{\cdot}66 \text{ H.P.}$$

We see, therefore, that while for continuous work and fair uniform loads a steam engine is the more convenient of the two, yet, for an occasional but heavy piece of work lasting but for a short time, a horse with its wide margin of strength is more convenient, and may even in some cases be the more economical.

Margin of Power in Horses.

When a race-horse is going with its rider at the rate of say 15 miles per hour, it will be moving at the rate of 1,320 feet per minute or 22 feet per second. Now, a horse going at the rate of 22 feet per second may take a jump over a hurdle 5 feet high. Let the total weight of horse and rider be 1,600 lb. Now in jumping a hurdle 5 feet high, the whole weight is only raised about 4 feet, as the horse folds its legs up in getting over, but the 4 feet in height is reached in one-half second at most, or at the rate of 480 feet per minute. Thus we have—

$$\frac{1{,}600 \times 480 \text{ feet}}{33{,}000} = 23{\cdot}2 \text{ H.P.}$$

Thus we see that for a short time, i.e. one-half second, a horse can develop at the rate of 23 times its normal power. This is a much greater margin of power than any steam engine can be expected to have, and a horse could not do it for a complete second.

We have shown how a horse can even for long periods pull against an intermittent load of 330 lb. and lift up this weight at a rate of 250 feet per minute. We have also seen how if the load be 660 lb. 2 horses can lift the same load at the same rate of speed. We will now consider how one horse can be enabled to lift a heavy load of say 660 lb. This can only be done by lifting it at a lower, and proportionately lower, rate of speed. It will be seen that merely letting the horse go more slowly does not answer, for, if it can only pull against 330 lb.

Fig. 4.

it does not matter how slow it goes, it does not give it any bigger pull. True, it may jerk a much heavier load up for a second, but that is very different from a regular continuous pull. We must therefore arrange so that the horse goes at its best regular speed, say 250 feet a minute, and that the load is raised at a speed as much slower in comparison with the speed of the horse as the load is heavier than the 330 lb.; thus, if the load be doubled to 660 lb., then it must go up half as fast or 125 feet instead of 250 feet a minute.

This is accomplished by fixing a drum or pulley at the top of the warehouse with two different diameters on it, as shown in Fig. 4. The larger of the two, on which the horse rope is coiled, is double the size of the smaller one, from which the load is suspended and lifted. For

Differential Pulleys.

every foot then that the load is lifted the horse moves 2 feet, and then has to resist only the same strain as before.

Assuming then that it is a continuous pull, we must take our standard 33,000 foot-pounds as our horse-power, the $\frac{33,000}{250}$ feet per minute=152 lb. weight which can be raised.

Again, $\frac{33,000}{600}$ lb. weight=55 feet per minute at which one horse can raise the weight.

The student should now have the idea well fixed in his mind, viz. that 33,000 foot-pounds equal one horse-power, and that the number of foot-pounds, 33,000, when divided by any *weight*, will give the speed at which the weight can be lifted in feet per minute. And second, that when divided by any *speed*, in feet per minute, it will give the weight in pounds which one horse can lift at that speed.

CHAPTER II

Natural Forces

BEFORE dealing with the design and use of steam engines, the student must get one other idea firmly fixed in his mind. It is this: There is no possible mechanical means by which power can be created; power cannot even be increased, excepting by a corresponding decrease of something else; thus, what is gained in one direction, is lost to exactly the same extent in some other.

No Creation of Power.

Many years ago the writer had the opportunity of hearing an address to students given by the late Sir William Siemens, a man of great scientific eminence in engineering and electricity. Speaking of inventions he said: "See first that your idea is a good one, i.e. that what you propose to do is worth doing, and not only that, but that it is better worth doing as you propose than it is now done. Then make sure that it is in accordance with the laws of Nature, and make yourself acquainted with these laws in order that you may know. Having made sure of these two things, then, let no labour however hard, no difficulty however great, no patience however long, deter you from doing what you wish."

Siemens' Dictum.

Now of all the laws of Nature these are most clear and decided. "We cannot create one atom of matter," "we cannot create one single unit of power," but we can modify and transform matters in a thousand ways, and we can utilize many great forces and sources of power which already exist. We are all familiar with the common crowbar or lever, and it may be objected that a man's power is much increased by its use, but it is not so. True, a man's ability to move a heavy weight is secured by the use of the lever, but his real strength or power is in no way greater than it was. The subjoined sketch shows the man raising a heavy stone. Here is a weight of 500 lb., which is much beyond the man's power to

move. He gets a lever 6 feet long, places one end under the stone, puts a fulcrum (as we call it) at *a*, one foot from the end of lever, and then presses with a force a little over 100 lb. on to the other end of the lever, and

The Lever.

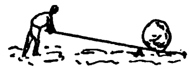


FIG. 5.

the stone is lifted say three inches high; but in order to do this, the man has had to press down with a force of 100 lb. for a distance of 5 times 3 inches, or 15 inches in all. Now 500 lb. × 3 inches or ·25 of a foot = 125 foot-pounds, which is the work done on the stone, and 100 lb. × 15 inches or 1·25 of a foot = also 125 foot-pounds, which is the power exerted by the man. The man's power then has not been increased but only utilized. Further, the man has obtained his strength from the food he has eaten, which has built up his body and formed his muscle and tissues; these have been wasted by the work he has done, and the exact equivalent of the work he has done must be made up by fresh food, or by a loss of the tissues of his body and in reduction in his weight.

Transformation of Energy.

It is well known that the harder a man works the more food he needs, hence the familiar practice of those who do no work, of taking a long walk or doing some violent physical exercises in order to get an appetite for food. In moving the body in walking, especially uphill, or in lifting dumb-bells, real work is being done, the body's tissues are being dissipated and the body becomes lighter owing to transference of tissue into work, but no power has been created, only natural forces transformed.

We wish to especially emphasize this truth because, not only in the past, when men were ignorant of Nature's laws, but even now, after so many generations of education, we find that year after year, month after month, almost day after day, patents are being applied for, for most absurd inventions—man engines, power multipliers, self-moving motors, and all sorts and varieties of perpetual motions. Some of these designs of perpetual motion seem at first sight very plausible upon paper, and it is not to be wondered at that those without precise scientific

Perpetual Motion Fallacies.

knowledge are deceived by them. One of the most deceptive types is shown in the sketch below (Fig. 6); here a wheel revolving upon its axis has a number of pins, *b*, *b*, *b*, *b*, *b*, *b*, *b*, *b*, from which pins are suspended rods, carrying weights at their outer ends. There are also an equal number of pins nearer the circumference of wheel at *c*, *c*, *c*, *c*, *c*, *c*, *c*, *c*, against which the rods can lean when in the positions *d*, *e* and *f*. When in this position the rods form radii to the wheel, while on the other side, as at *g*, *h* and *i*, they hang down freely and the weights are in this position much nearer the centre of the wheel. In the position shown in the sketch the weight *g* and *k* exactly balance each other, while *d*, *e* and *f*, though not more numerous than *h* and *i*, are

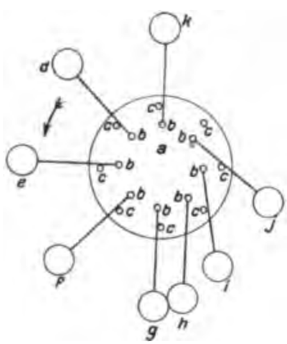


Fig. 6.

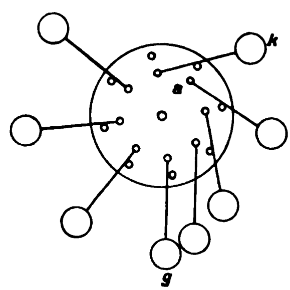


Fig. 7.

much farther from the centre, and it is quite evident that such a wheel must turn round in the direction of the arrow, and if the rods and weights would continue to assume these positions, then the wheel would continue to revolve and would exercise power proportioned to the weight of balls and length of the rods; but it is just here where the fallacy of the inventor comes in. If his invention ever reaches the testing stage, he places the balls in the position shown and in so doing has expended some power himself in lifting the balls *j* and *k* into this position. Until this outside power has been expended the wheel would revolve, but would soon slow down, when the balls and rods would take their natural position of equilibrium, as shown in the illustration (Fig. 7). Here it will be seen that the weight *k*, instead of being vertical is on the right

hand of the figure, as it cannot fall over to the left until after it has reached the position shown in the dotted lines, and by this time the weight *g* will have passed the centre line and will have started ascending again. Thus there will never be more than three weights in the extended position helping the revolution, and five in the inferior position hindering the revolution, this greater number and less leverage, exactly counterbalancing the smaller number and greater leverage. There have been many variations of this type of power machine having the same fundamental defect. No piece of mechanism can ever move itself.

Man Engine. About the year 1860 a very clever mechanic constructed what he called a man engine, by means of which he claimed one man could exert the power of 10 men. This mechanic had a little mechanical knowledge and much ingenuity. He knew something of the laws of levers, how that if the distance from the fulcrum *a* to *b*,

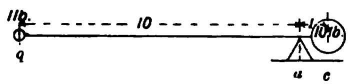


Fig. 7A.

is 10 times the distance from *a* to *c*, then a weight of 1 lb. at *b*, will balance a weight of 10 lb. at *c*. So by a complicated arrangements of lever cranks and belts, he managed to do what he rightly said a great many inventors had attempted and failed, i.e. he managed to make the short arm of one lever work the long arm of another, and thus (he said) increased his power tenfold. His method of doing this is shown in the following diagram.

The man worked the lever *a*, to the short end of which was attached a connecting rod which worked the crank-shaft *B*. Upon one end of the crank-shaft was a fly-wheel, and on the other end a sprocket wheel, a corresponding sprocket wheel being at *D*. Upon one of the links of the sprocket band, going round these two wheels, was fixed the female part of a universal joint. Working through a hole in the ball of this universal joint was the long end of another lever. This lever had a long vertical joint at *E*, thus enabling it to move or bend sidewise, while retaining its rigidity in a vertical direction. When the man presses down the lever *a*, the

sprocket wheel C revolves in the direction of the arrow, the long end of the lever F is raised, passes over the top of sprocket wheel C, down the other side round under the bottom of wheel D and back to its position as shown, thus in its progress

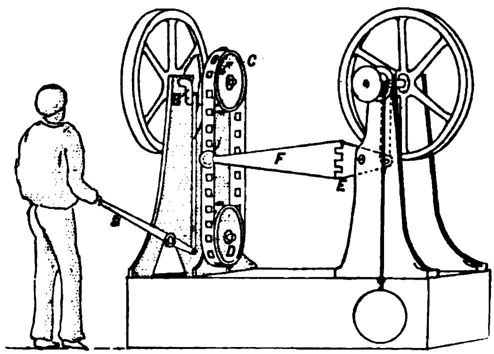


FIG. 8.

demonstrating the fact that, "the short arm of one lever can be made to move the long arm of another through a vertical distance equal to that of the long arm of the first lever." The short arm of the second lever was jointed to a connecting rod which turned a crank, on the shaft of which was a drum with a rope coiled round it from which was suspended a heavy weight. This weight was drawn up by the machine when a man worked the lever *a*. The levers were each 4 to 1, and thus the inventor claimed that as the man gained 4 times the power with one lever, and its short arm worked the long arm of the other, the real gain was $4 \times 4 = 16$, so that in claiming to enable one man to do the work of 10, he was really well within the mark.

The inventor constructed a machine in his spare time, and his demonstration was so plausible that two lawyers and a prominent banker formed a small syndicate to work his invention as a commercial venture, their first object being to form a big company with a large capital for the purpose. To help them in this object a public demonstration was advertised in an important engineering centre, and to this demonstration the writer went with an engineering friend, examined the model, and saw the wonderful demonstration, how with a very slight pressure on the end of lever *a*, the

great weight rose majestically from the ground amid the applause of the public. The inventor looked triumphant, when the writer's friend, who had had the advantage of a technical training, approached the inventor and said, "Do you sincerely wish us to believe that when you work that lever with the force of one man, that you really have the power of ten men on the pulley which carries up the weight?" "Certainly I do," said the inventor; "I have just demonstrated that fact. Can you not see for yourself?" "Ah,

The Inventor's Enlightenment.

then," said the engineer, "I very much wonder why you take the trouble to work the lever at all, why do you not connect the end of the machine to the beginning? then it would work itself and you would have the power of nine men to spare." It was pathetic to see the change which gradually came over the face of the sanguine inventor as a new and terrible conviction crept into his brain. The exhibition closed that night and he went home "a sadder but a wiser man." Nothing further was heard of the great invention or the big company for working the same.

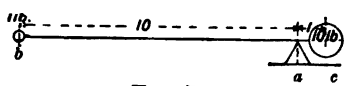


Fig. 7A.

A knowledge of the fact that speed and power cannot be both gained by any mechanical means at the same time, would have saved the inventor all his trouble. It was true that with a pressure of 10 lb. on the end of lever he raised a weight of 100 lb. on the drum, but only at exactly $\frac{1}{10}$ of the speed, and he would have done it much more simply by one plain lever, like that shown in illustration (Fig. 7A), or if it was needed to raise weight continuously, then, by a simple handle on a shaft, and a small diameter of drum on same shaft like a well tackle (Fig. 9). Here if the man's hand goes through 10 times the distance that the rope does, a weight 10 times as heavy as the pressure on the handle can be lifted by the rope and no more.

An inventor of a bogus machine is not always so honest as the man we have described, and one or more such swindles

Keely Motor.

are always before the world; the inventor, or more probably the owner of them, deliberately gulling an ignorant but greedy public. One of the most remarkable

and notorious is that known as the "Keely motor," and cleverly described and burlesqued in the *Scientific American* some years ago :—

"There are a number of pipes, wheels, pulleys, rods, belts, levers, cocks, cams and cogs visible, besides, it is darkly hinted, a vastly greater number of the same sort of thing under the floor and back of the partition. In front of the motor is Mr. Keely's office, in which there is a large slot. The shareholders drop their money in the slot and Mr. Keely looks out and watches them walk away.

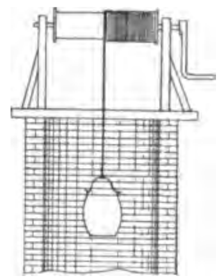


FIG. 9.

"Naturally he is sometimes called upon to explain the working of his machine by some doubting shareholder. And—so says the *Scientific American*—this is how he does it. He receives his victim all smiles, and, conducting him into the presence of the machine, he says: 'You see, my friend, the way we operate the motor is this. Taking hold of the lever, we pull it towards us. This causes the small flip flap you see there to be withdrawn, allowing the flibber-snatcher to fall into its place on the ramrod. As soon as this happens, it acts directly on the hatchway and the slam-bang, causing them to make a half revolution and start the get-up-and-get motion of the flunkee-flopper, which in turn communicates its energy to the button-hook and the wapper-chock.

"'After these things have run for about five minutes, they cause the jig-gag valve to turn, and the asthmatic gas flows through the pipe to the cylinder, and gives the wiggle motion to the gilder-fluke—that's the point we are striving after—the wiggle motion of the gilder-fluke. Why, my dear sir, without the wiggle motion of the gilder-fluke you wouldn't think of putting your money into the motor. But with it, sir, we are—eh, another share? All right, come into the office and I'll have it made out for you inside of a minute.'

"Experience has shown that Professor Keely has been much more successful in the mechanical manipulation of the shareholders' money than in the management of his motor. Taking hold of the middle of a bill of any denomination with the thumb and forefinger of each hand, he holds the end of

the bill towards his person. By a dexterous movement of the fingers, he causes the bill to fold across the centre; repeating the process, he has it reduced to the proper compass for wadding into his pocket-book, which is the next movement. This most ingenious gentleman, Don Keely, then places the purse in his right hand trousers pocket, and smiles quietly."

The above is of course a burlesque, but not a very gross exaggeration of the way in which some unscrupulous persons are always trying to delude the public, which, alas! so long as it is ignorant, will always be deluded.

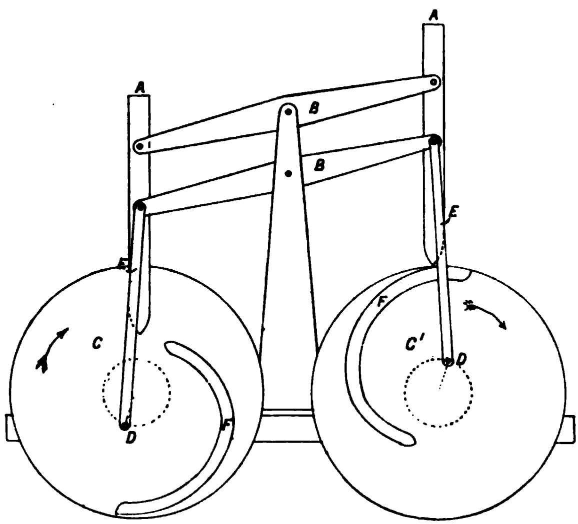


FIG. 10.

We have gone thus fully into the matter of so-called perpetual motions and motive power engines so that the student may be thoroughly convinced that any attempt to create power mechanically must necessarily fail. We can in a thousand ways utilize natural forces, modify their action, and transform energy into heat, and heat into motion; we can also effect marvellous chemical transformations, but we can create neither matter nor power. There are also severe natural limitations to our utilization of natural forces. About the year 1860, the author, having at that time a smattering of knowledge about magnetism, invented what he then believed to be a magnetic engine. It was not to create but to utilize

the power which magnets undoubtedly have to attract masses of soft iron to themselves, often with considerable force. This motor is shown in Fig. 10.

Magnetic Engine. A and A' are two bar magnets suspended each at the ends of a pair of brass beams, B, B, so that they always move up and down in a vertical line and balance each other. C, C are 2 discs of brass with crank pins projecting from their face at D, D. The crank pins are connected to the beams by the connecting rods, E, E, also of brass, as are the beams. The discs revolve in the direction of the arrows. Upon each disc is secured a soft iron armature, F, F, of such a shape that as the disc revolves and the magnet comes down, it always keeps a very slight distance away from it, until a half revolution is accomplished, during which it draws the magnet down to its inferior point and then breaks off. In the left hand disc the armature has just left the magnet, which is now free to go up again, while the right hand armature is just commencing to draw its magnet down. Thus, alternately, each end of the beam is drawn down, and the discs are rotated in the direction of the arrows. The weight of each armature is of course balanced by suitable counterweights behind the disc. From all appearance it looked as if such a machine should develop power in proportion to the pull of the magnets and the distance through which they moved. Unfortunately for this theory, when made, the machine absolutely refused to work with any surplus power whatever, and at first this seemed inexplicable. The magnets did, without question, exert a pull of about ¼ lb. each on the armature, and, conversely, the armature did pull down the magnets with this force, first on one disc and then on the other, thus tending to cause the discs to revolve. The discs nevertheless remained stationary, and it was not until years afterwards, when the principles governing the construction of dynamos began to be understood, that the cause became manifest, viz. that not only does a magnet pull in the familiar direction of its length, attracting the armature to it, but that there is a real friction or resistance to the movement of the armature sideways, even when not touching the magnet; this resistance is less than the direct pull, but it varies in intensity with it, and, as the distance from one end of the armature to the other is much greater than the direct distance pulled down, the smaller resistance through the greater length, balanced the stronger attraction through

the smaller distance, and the result was equilibrium. Many other magnetic engines have been made both before and since that date and of widely different designs, but none have become sources of power.

The above limitation of our powers does not prevent the engineer from utilizing forces which exist. Owing to various natural causes, the atmosphere is frequently in a state of motion, and we make use of this motion to work windmills and propel ships on the water. Natural forces have made, and are always making, immense stores of water in elevated situations; this water in descending again to sea level, can, and is, made to do work for us through the means of water-wheels and turbines. These methods of utilizing natural forces are familiar, and have been familiar to all for ages. Among the greatest of our natural forces is the heat, or the possibility of producing heat, stored up for us in immense quantities in the timber and coal of the world, and it is with this heat we shall have principally to do in our future chapters, but it may be well to say here that while creation of matter or power is impossible, yet there may be many forces in nature that we have not yet discovered, and nothing should prevent us searching in every direction for the means by which these forces may be found and utilized.

Electricity. This has rightly been called the age of electricity, and marvellous discoveries in connexion with this great force have been discovered of late years: the electric telegraph, the telephone, the phonograph, electric lighting on a commercial scale, and lately electric telegraphing without wires, or as it is called wire-less telegraphy, but nearly all these great uses of electricity depend upon other sources of power: windmills, water wheels, or steam or gas engines, all of these being either natural or artificial methods of utilizing heat. The more electricity is used the more steam engines are required in order to generate the electric currents by means of electric generators or dynamos as they are called, and so far as we can tell, such engines will be required in still greater numbers for many years to come. It is, however, far from impossible, and by no means improbable, that, just as currents of wind are naturally produced, and can be utilized by us, just as immense volumes of water are always being taken up by natural means and stored in vast elevated lakes without human

effort, so there may be, and probably are, immense currents of electricity always circulating round the earth, and vast stores of electric energy in the atmosphere and the earth, which stores have not yet been found and utilized. How far electric disturbances may be responsible for abnormal tidal waves and destructive earthquakes, we give no opinion, but that electricity is responsible for destructive thunder storms in the atmosphere we all know, and, just as the uncontrolled emptying of natural reservoirs of water produce floods, disaster and ruin, so do the electrical discharges over which as yet we have no control. It is quite conceivable that at some time, perhaps not far distant, we shall have discovered the means by which we may reach and control these natural electrical forces and thus make ourselves independent of wood, coal and oil as a source of heat, because we have succeeded in harnessing these powerful natural forces. Then, instead of them producing disasters in earthquakes and atmospheric storms, they may be our obedient servants to do our work in the way we direct.

Until we have done this we must continue to depend upon our power to produce artificial heat, and we must try to understand what heat is, and how it acts as a source of power.

Heat.

We are all familiar with the heat we get from the sun, and from burning fuel. Many may at some time have picked up a bullet, which, just shot from a rifle, has flattened itself against an iron target, and dropped it suddenly because it was so hot; some few may have seen a blacksmith hammer a piece of cold iron until it became red hot, and there are probably few engineering apprentices who have not made, or seen, the simple apparatus for producing heat, and lighting a pipe shown in the sketch (Fig. 11), here *a* is a small cylinder, bored out perfectly true and smooth, and at least 8 decimetres long, *B* is a piston and rod fitting truly and easily in the cylinder. Into the lower surface of piston is filled a piece of tinder or "amadou," which takes fire easily. If now the piston be inserted at the upper end of cylinder, and a smart blow be given on the cap *C* with sufficient force to send the

Transference of Motion into Heat.

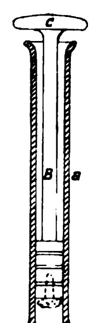


FIG. 11.

piston nearly to the bottom of the cylinder, a great increase in the pressure of air will take place, accompanied, as it always is, by an increase in the temperature. This will be so great as to set the tinder on fire, which, when the piston is withdrawn, will be found all in a glow.[1] Here we have three different examples, not of the creation of heat, but of the transformation of motion into heat. In the first case the bullet had a tremendous velocity or motion given to it, which motion was suddenly arrested as if it were by one blow, and all the stored-up energy in the bullet was transformed into heat. Some of the heat went into the target, and was dissipated in its mass, but the bulk of it served to raise the temperature of the bullet till it became too hot to hold, and, had the velocity of the bullet been great enough, on being suddenly stopped it would have melted, or have been transformed into vapour. With regard to the second illustration, instead of one great blow, the smith gives the iron a great number of small blows in rapid succession, and his energy transformed into the motion of the hammer is converted into the heat of the iron. A similar thing takes place with the pipe-lighter; it is the force of the blow which is struck on the piston head, which compresses the air in the cylinder, and reappears in the form of the heat which sets the tinder on fire. We shall deal later with some of the laws which regulate these phenomena, but it is enough to learn now that motion can be transformed into heat, and heat into motion.

Bullet and Target.

Heat a Mode of Motion.

Heat has been aptly called a "mode of motion."[2] Heat causes particles of matter of all kinds to tend to pass from the solid quiet stage into a state of motion, such motion being greater in proportion to the heat. So to take a familiar illustration, a lump of ice is solid and quiet; subject it to heat, when it first becomes saturated with the heat, and then melts into water. In this state its particles are much more free to move; it is now in what we call the fluid state, in which every atom is in a state of constant motion. Let still more heat be added, and the atoms become more and more agitated,

Ice, Water, Steam.

[1] The heating of a bicycle pump when the tyres are being inflated is due to the same cause, but as the pressure attained is not so great the heat is less.

[2] Those who wish to go fully into the subject should read works by Professors Tyndall and Clark Maxwell on heat.

strike against each other, rebound, and fly off from each other in the form of vapour, or steam as we call it. In this state they occupy a much larger space than the liquid did, a cubic inch of water converted into steam of 5 lb. absolute pressure, occupying a space 4,527 times as great as it did in the liquid state and at atmospheric pressure, i.e. practically 15 lb. per square inch, the steam occupies 1,610 times the space required for the water.

If now the steam is allowed to enter a vessel of suitable size, in which there is an aperture for the escape of the air, the steam will first mix with, and then expel all the air and completely fill the vessel. If the aperture which admitted the steam and allowed the escape of the air be now closed, we shall have one cubic inch of water filling a space of 1,610 cubic inches, and the pressure within and without the vessel will be the same. If now we inject into the vessel a tiny jet of cold water, the steam will instantly be cooled and return to its state of water, occupying again only its one cubic inch of space plus the small quantity of condensing water. There is now no pressure inside the vessel, but the outside pressure of the atmosphere, 15 lb. to the square inch, will still press on all sides. The vessel has a capacity of nearly one cubic foot, and if it be rectangular will present on its 6 sides an area of 850 square inches. Now 850 × 15 lb.=12,750 lb., considerably over 5 tons outside pressure, and unless it be very strong the vessel will collapse and crumple up.

It is not necessary to inject the cooling water, though that is the best way. Cold water applied outside will do, but a greater quantity will be required.

Condensation Experiment. A simple experiment can easily be made to illustrate the above statement. Take an ordinary biscuit box, let the lid be soldered on tightly, leaving a small filling hole into which a good cork must fit air tight. Now put a little over a cubic inch of water into the box and boil it over a spirit lamp or bunsen burner till the steam comes freely out of the hole in top; then turn out the flame and at the same time insert the cork firmly. Let a cloth soaked in cold water be now applied to the outside, when almost immediately the box begins to collapse, and then suddenly crumples up into a shapeless mass.

It is evident we have here a natural force of great power,

which, by suitable means may be utilized to do useful work.

The Steam Cylinder. Instead of a tin box let us have a long cylinder A, shown in Fig. 12. If instead of a fixed lid, we have a lid fitting air tight within the cylinder, and free to slide up and down within it, such a lid would be called a piston and is shown at B. Let us suppose now that the cylinder is filled with steam at atmospheric pressure, and that the lid or piston is at the top as shown. If now a small jet of cold water be sent in through the cock C, the steam will be instantly condensed into water, and the weight of the atmosphere pressing on the piston B, will drive it down with a force depending upon its area in square inches multiplied by 15. Thus assuming that the area of the piston is 1,000 square inches, then the pressure on its surface would be $1{,}000 \times 15 = 15{,}000$ lb., or nearly 7 tons.

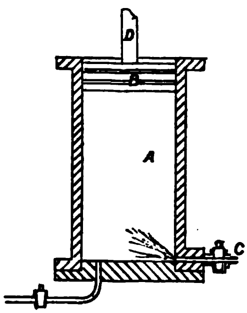


FIG. 12.

It will be easily seen that if the piston rod D were attached to one end of a beam working on a centre, that the descending piston, going down with this force would be able to lift a weight of over 6 tons hanging on the other end of the beam. Let Fig. 13 show such an arangement where A is the cylinder shown in section, B the piston, C the injection cock, and D the connexion between the piston and the oscilating beam F. At each end of the beam it will be seen is a segment of a circle, struck from the beam's centre, a linked chain attached to G and G' respectively, connects the piston with the beam at one end, and the weight E with the beam at the other end. It is obvious now that when the cock C is opened, the cold injection admitted to the cylinder, and the steam condensed, the weight of the atmosphere will force down the piston, and (if it be not too heavy) the weight E will be raised. When the cock H is opened to let out the water and C to admit more steam, the piston will be drawn up to the top of the cylinder by the descending weight E, but so far we have done no useful work.

Primitive Steam Engine.

The pressure on the piston is 15,000 lb., and the weight may be a little less than this, say 14,000 lb. This weight in descending can do much more than draw up the piston; in

its descent it could raise another weight a little less than itself, say 13,500 lb. Now let us conceive that a rod from the bottom of weight *E* goes down a mine or pit, from which we want to draw water. Let a rigid rod be the connexion between the weight *E* and a pump plunger working in a suitable barrel or pump case with the necessary valves. Assuming that the pump case is full of water when the weight and plunger pole descended, the water would be displaced, and could be conveyed by pipes to the surface.

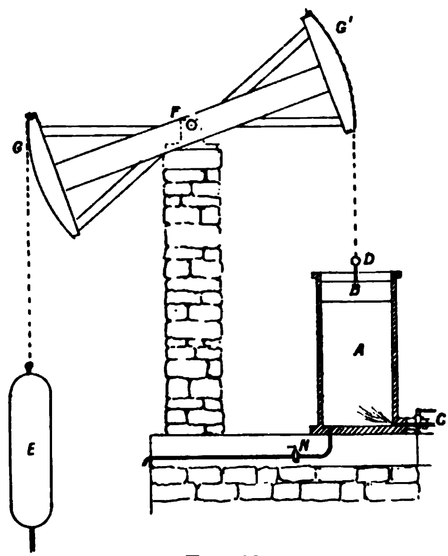


FIG. 13.

We will now consider how much water could be forced up, or raised as we call it, at each stroke. We have seen that the descending weight being 14,000 lb., the weight of the column of water may be 13,500 lb. We will assume that the height to which the water has to be raised is 500 feet. Now a column of water 30 feet high and one square inch in section, balances the weight of the atmosphere on one square inch; the atmosphere though so light, making up its lightness by great depth (being considerably over 5 miles high), makes a pressue of 15 lb. on each square inch.[1] Thus one lb. pressure equals a column of water 2 feet high. As then the height to which the water has to be lifted is 500 feet, this figure divided by 2 feet will give the pressure in lb. of each square inch of the column, viz. 250 lb.; our total weight of water being 13,500 lb., this number ÷ 250 will give us the number of square inches which the cross section of the pump

Weight of Atmosphere.

[1] The figures are not exact, as the weight of the atmosphere constantly varies, and with it the height of a column of water supported by it, but for all practical purposes we take atmospheric pressure at 15 lb., and 30 feet high as that of column of water balanced by it.

plunger should have, and 13,500 ÷ 250 = 54 square inches. As a cylindrical pump plunger and case is by far the form most easily made, we have now to find what *diameter* the pump plunger must be.

The Plunger Pump.

It is evident from the drawing below (Fig. 14), that a square inch is larger in area than a circular inch, the subtraction of the shadowed corners *a*, *a*, *a*, *a*, reducing it by nearly one quarter of its area. To find the exact size of square which equals the area of a given circle is one of the well-known mathematical impossibilities, but we know to the nearest ten-thousandth, and that is quite near enough for all practical purposes. The circle that can be contained within a square *a* in Fig. 14 is ·7854, i.e. a little over ¾ of its area; thus, any number of circular inches × ·7854, will give the number of square inches which correspond to the same total area. A square 10 inches each way, or as we call it, 10 inches square, as will be seen by Fig. 15, has one hundred squares in it, of which 78·54 are within the circle and 21·46 without the circle.

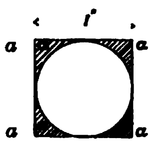


FIG. 14.

Square inch and

Square Foot.

On examining the square carefully it will be found that there are but 60 complete squares within the circle and 12 without, there remaining 28 which are cut by the circular line, and thus belong partly to both, but in very irregular quantities; the portions of the 28 which lie within the circle, all added together, make 18·54 which, added to the 60 complete ones, make up the 78·54, and the portions of the 28 which lie without the circle, added to the 12 complete ones, make up the 21·46, and 78·54 + 21·46 = 100. If the circle was 20 inches diameter, then 20 × 20 = 400, and this sum × ·7854 = 314·16, being the number of square inches contained in the circle.[1]

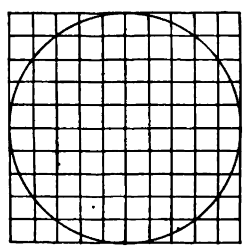


FIG. 15.

[1] *Note.*—There is a curious relation between the diameters, areas, and circumferences of numbers, which may as well be noted here, as a knowledge of it will be useful later on. Every one knows that it is about three times as far to go round a circle, as across it. In fact, it is a little more than three times, exactly 3·1416. Now ·7854 (the factor which gives the relation between a square and the circle), if multiplied by 4 = 3·1416, and the diameter of any circle multiplied

Conversely, to find the area of a square, which encloses the area of a circle of a given diameter, say 20 inches diameter, we must find a factor which is as much larger than 1, as ·7854 is smaller; this latter number then divided into 1, will give us the factor $\frac{1}{\cdot 7854} = 1\cdot 2733$, and any number of square inches multiplied by 1·2733 will give us a number, the square root of which will be the diameter of a circle which will contain that number of square inches. Thus to take the number we have been dealing with, 314·16 square inches, then $314\cdot 16 \times 1\cdot 2733 = 400$, the square root of which is 20 inches, this will be the diameter of the circle. What we have been wanting to find is the diameter of a pump plunger which has an area of 54 square inches. Thus $54 \times 1\cdot 2733 = 68\cdot 75$ circular inches, and the square root of this number, viz. 8·29, will give us the diameter of the ram or plunger we require. This is the largest diameter we could make the plunger, but in order to leave a little margin, and to reduce it to even figures, we will make the plunger 8 inches diameter, which will give it an area of 50·26, say 50 square inches. We have found that the weight of a column of water 500 feet high gives a pressure of 250 lb. per square inch, then $50 \times 250 = 12{,}500$ lb., the full weight of water to be lifted at each stroke.

Speed of Water in Pumps. The speed at which water may be lifted by a plunger pump is found to be about 150 feet per minute, and, as on the return stroke no water is being lifted, the average speed at which the water is being lifted is 75 feet per minute. Now we have 12,500 lb. to lift at an average speed of 75 feet per minute, $\frac{12{,}500 \times 75}{33{,}000} = 28\cdot 4$H.P.,

by 3·1416 will always give the correct circumference. Now any circle whose diameter is 4, of any factor, i.e. 4 mm., 4 inches, 4 yards, or 4 miles, when multiplied by 3·1416, the result will be not only its circumference, but also its area, thus, $4 \times 4 \times \cdot 7854 = 12\cdot 56$ area, and $4 \times 3\cdot 1416$ also $= 12\cdot 56$, both circumference and area. Further, if the diameter be the double of 4, i.e. 8, then the area will be double the circumference. If the diameter be 4 times 4, i.e. 16, then the area will be 4 times the circumference. If the diameter be 8 times 4, i.e. 32, then the area will be 8 times the circumference, and so with any similar multiple of 4; thus 64 inches which is 16 times 4, has an area 16 times its circumference. This curious factor, 3·1416, is called by the Greek letter Pi, and is much used in algebraical calculations. Also with regard to spheres, the square of the diameter multiplied by Pi gives the surface area.

What is Pi?

and this is the work being done by one end of the beam in Fig. 12. The other end of the beam is connected with a piston, whose area is 1,000 square inches, and working with an atmospheric pressure of 15 lb. to the square inch, we have 15,000 lb. pressure, the speed of the piston is the same as that of the plunger, viz. 75 feet per minute; thus $\frac{15{,}000 \times 75}{33{,}000} = 34 \cdot 09$ H.P., the power the engine's cylinder can give out. The difference between 28 H.P. and 34 H.P. being required to overcome the friction of the parts, and to furnish the necessary margin of power to ensure movement, it being evident that if there were perfect balance, there would be just equilibrium of forces, but no motion.

CHAPTER III

Early Types of Engine

The knowledge of the nature and powers of steam had arrived at the stage described in the last chapter when Newcomen in 1705 invented and made what, so far as we have been able to learn, was the first steam engine which did any useful practical work. This engine is shown in the diagram below, and, if what has gone before has been properly assimilated, its construction and method of working will be clearly seen from the sketch.

Newcomen's Engine.

Here we have the boiler *A* which produced steam at atmospheric pressure only. The steam cylinder *B* is fixed immediately above it and communicates with the boiler by the short pipe shown, in which was a cock which was opened and shut by hand. *C* is a tank of cold water which is connected by a pipe to the bottom of cylinder to give the injection water. The cycle of operations is similar to that described in connexion with Fig. 13, i.e. the cylinder having been filled with steam and the piston being in the position shown, the injection cock *D* is opened, the cold water rushes in, in the form of spray, the steam is at once condensed into water, thus producing a vacuum, and the weight of the atmosphere on the top of piston *E* forces it down to the bottom of cylinder, at the same time raising the

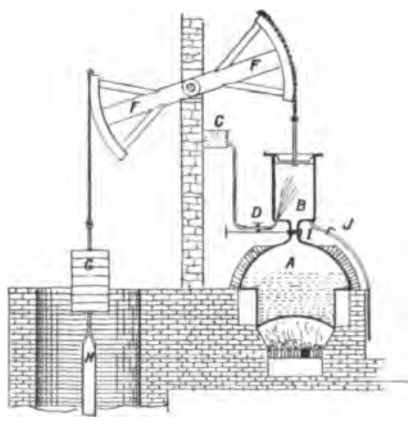


Fig. 16.

other end of beam *F* and with it the weight and the pump rods *G* and *H*. At the bottom of the pit shaft was the pump, operated by the rod *H*. When the piston was at the bottom of the stroke and the pump rods at the top, the injection cock *D* was closed, the cock *J* opened to allow the water of condensation to drain away. This cock was then closed and the cock *I* opened to admit steam. The weight then of the weight *C* and the pump rod *H* caused that end of the beam to fall and the steam piston to rise, when the operations are repeated.

For some years after its introduction the cocks and valves were operated by hand and the working of the engine was necessarily slow. Tradition tells us that a boy named Humphrey Potter, engaged for the purpose of working the valves, and longing for freedom and time in which to play at marbles, devised and fixed a system of cords and levers by which the cocks were opened and closed by the movement of the engine beam. This crude apparatus of his was soon improved upon and became the precursor of the plug tree and cataract motion still in use on many pumping engines.

Though this type of engine remained in use over seventy-five years, yet, on looking at the diagram (Fig. 16), it will be seen at once that it only does any work by means of the steam, during one-half of a complete cycle of operations, i.e. a complete up and down stroke, and that it is the down stroke of the piston due to the pressure of air upon its upper surface, and the vacuum below it, when the work is really done. If now a cover could be put on the cylinders and inlet and outlet cocks suitably arranged, the vacuum might be formed on both sides of the piston and thus from the same size of cylinder twice the power could be obtained.

The Stuffing-Box.

Two difficulties however arise; first, a hole must be made in the cover through which the piston-rod has to work, and this hole must be made steam tight. This difficulty is overcome by what is known as a "stuffing-box"; this is shown in detail in Fig. 17 below. *a* is the piston-rod attached at one end to the piston and at the other end to the beam (through suitable connexions); *b* is the centre part of the cylinder cover, and round the hole in it, is made the stuffing box *c*, *c*. This box was fitted with greased tow or hemp; just fitting within this box was the flanged cylinder *d*, *d*, called the "gland"; this rested on the greased hemp or

"stuffing," and by means of the nuts and bolts *e e*, was forced down into the box, thus causing the stuffing to press tightly round the rod and prevent the escape of steam.[1]

Watt's Invention.

The next important improvements in the method of using steam were made by James Watt, who, in investigating the working of a model of Newcomen's engine (Fig. 16), discovered that an immense waste of heat occurred by condensing the steam within the cylinder in which it was used. Metals are known to be very good conductors of heat. The subject of conduction and radiation of heat will be dealt with more fully later on, but now it is important to recognize the well-known difference in heat conduction, between say iron, lead and wood. An iron poker when made red-hot in the fire at one end will be much too hot to hold comfortably in the hand at the other end owing to the rapid conduction of heat from one end to the other, but a leaden rod or pipe of the same length may be melted at one end when the other is only slightly warmed, while a long stick of wood may be brightly burning at one end and may consume almost close up to the hand without sensibly raising the temperature. Wood therefore is almost a non-conductor, lead is a bad conductor, while iron is a very good conductor of heat.

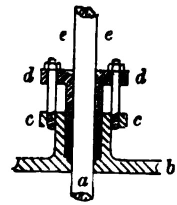


FIG. 17.

Conduction and Radiation of Heat.

Heat of Steam.

Now the heat of the entering steam at that of boiling water only is 212°, and it rises as the pressure rises, though not in the same proportion (*see* table in appendix, p. 359); at say 5 lb. above the pressure of the atmosphere the temperature is 228°, while that to which it is reduced when condensed to say 1 lb. is but 102°. When, therefore, the steam at the temperature of 228° entered the cylinder, it had first to raise the temperature of its walls and piston to its own, and in doing this a large proportion of the steam was condensed and therefore lost in the operation. Again when the steam had to be condensed, not only it, but also the walls of the cylinder had to be cooled before the vacuum could be made,

[1] In its essential features the stuffing-box remains to this day, but improved methods of packing suitable to high pressure steam will be dealt with later on under engine details.

it was thus cooled down again nearly to its low temperature of 102°, and all this had to be repeated at each stroke.

Now a steam engine is a machine for utilizing one of the great forces of nature, heat; any waste of this must therefore be avoided if it be possible. It may be well here to tell in Watt's own words how he dealt with this problem. Watt published this himself in a note to an article on the steam engine in Robison's *System of Mechanical Philosophy*, published in the year 1822. Though this note is a very long one, yet, as it contains the foundation and germs of nearly all the subsequent improvements which have been made in steam engine design, it is well worthy of study, but is not essential to be read at this stage.

Watt's Description of his Invention.

"My attention was first directed in the year 1759 to the subject of steam engines by the late Dr. Robison himself, then a student in the University of Glasgow, and nearly of my own age. He at that time threw out an idea of applying the power of steam engines to the moving of wheel-carriages, and to other purposes, but the scheme was not matured, and was soon abandoned on his going abroad.

"About the year 1761 or 1762 I tried some experiments on the force of steam, in a Papin's digester, and formed a species of steam engine by fixing upon it a syringe, one-third of an inch diameter, with a solid piston, and furnished also with a cock to admit the steam from the digester, or shut it off at pleasure, as well as to open a communication from the inside of the syringe to the open air by which the steam contained in the syringe might escape. When the communication between the digester and the syringe was opened, the steam entered the syringe, and by its action upon the piston raised a considerable weight (15 lb.) with which it was loaded.

"When this was raised as high as was thought proper, the communication with the digester was shut, and that with the atmosphere opened; the steam then made its escape, and the weight descended. The operations were repeated, and though in this experiment the cock was turned by hand, it was easy to see how it could be done by the machine itself, and to make it work with perfect regularity. But I soon relinquished the idea of constructing an engine upon this principle, from being sensible it would be liable to some of the objections against Savery's engine, viz. the danger of bursting the boiler, and the difficulty of making the joints tight, and also that a great part of the power of the steam would be lost, because no vacuum was formed to assist the descent of the piston. [I, however, described this engine in the fourth article of the specification of my patents of 1769; and again in the specification of another patent in the year 1784, together with a mode of applying it to the moving of wheel-carriages.]

"The attention necessary to the avocations of business prevented me from then prosecuting the subject farther; but in the winter of 1763–4, having occasion to repair a model of Newcomen's engine belonging to the Natural Philosophy Class of the University of Glasgow, my mind was again directed to it. At that period my knowledge was

derived principally from Desaguliers, and partly from Belidor. I set about repairing it as a mere mechanician, and when that was done and it was set to work, I was surprised to find that its boiler could not supply it with steam, though apparently quite large enough (the cylinder of the model being 2 inches in diameter, and 6 inches stroke, and the boiler being about 9 inches diameter). By blowing the fire it was made to take a few strokes, but required an enormous quantity of injection water, though it was very lightly loaded by the column of water in the pump. It soon occurred that this was caused by the little cylinder exposing a greater surface to condense the steam than the cylinders of larger engines did in proportion to their respective contents. It was found that by shortening the column of water in the pump, the boiler could supply the cylinder with steam, and that the engine would work regularly with a moderate quantity of injection. It now appeared that the cylinder of the model being of brass, would conduct heat much better than the cast-iron cylinders of larger engines (generally covered on the inside with a stony crust), and that considerable advantage could be gained by making the cylinders of some substance that would receive and give out heat slowly; of these, wood seemed to be the most likely, provided it should prove sufficiently durable.

"A small engine was therefore constructed with a cylinder 6 inches diameter, and 12 inches stroke, made of wood, soaked in linseed oil, and baked to dryness. With this engine many experiments were made; but it was soon found that the wooden cylinder was not likely to prove durable, and that the steam condensed in filling it, still exceeded the proportion of that required for large engines according to the statements of Desaguliers. It was also found that all attempts to produce a better exhaustion by throwing in more injection, caused a disproportionate waste of steam. On reflection, the cause of this seemed to be the boiling of water in vacuo at low heats, a discovery lately made by Dr. Cullen, and some other philosophers (below 100°, as I was then informed), and, consequently, at greater heats the water in the cylinder would produce a steam which would, in part, resist the pressure of the atmosphere.

"By experiments which I then tried upon the heats at which water boils under several pressures greater than that of the atmosphere, it appeared, that when the heats proceeded in an arithmetical, the elasticities proceeded in some geometrical ratio; and by laying down a curve from my data, I ascertained the particular one near enough for my purpose. It also appeared, that any approach to a vacuum could only be attained by throwing in large quantities of injection, which would cool the cylinder so much as to require quantities of steam to heat it again, out of proportion to the power gained by the more perfect vacuum; and that the old engineers had acted wisely in contenting themselves with loading the engine with only 6 or 7 lb. on each square inch of the area of the piston.

Bulk of Steam.

"It being evident that there was a great error in Dr. Desagulier's calculations of Mr. Beighton's experiments on the bulk of steam, a Florence flask, capable of containing about a pound of water, had about one ounce of distilled water put into it; a glass tube was fitted into its mouth, and the joining made tight by lapping that part of the tube with pack thread covered with glazier's putty. When the flask was set upright, the tube reached

down near to the surface of the water, and in that position the whole was placed in a tin reflecting oven before a fire, until the water was wholly evaporated, which happened in about an hour, and might have been done sooner had I not wished the heat not much to exceed that of boiling water. As the air in the flask was heavier than the steam, the latter ascended to the top, and expelled the air through the tube.

When the water was all evaporated, the oven and flask were removed from the fire, and a blast of cold air was directed against one side of the flask, to collect the condensed steam in one place. When all was cold, the tube was removed, the flask and its contents were weighed with care; and the flask being made hot, it was dried by blowing into it by bellows, and when weighed again, was found to have lost rather more than 4 grains, estimated at 4½ grains.

"When the flask was filled with water, it was found to contain about 17½ ounces avoirdupois of that fluid, which gave about 1,800 for the expansion of water converted into steam at the heat of boiling water.

"This experiment was repeated with nearly the same result; and in order to ascertain whether the flask had been wholly filled with steam, a similar quantity of water was for the third time evaporated; and, while the flask was still cold, it was placed inverted with its mouth (contracted by the tube) immersed in a vessel of water, which it sucked in as it cooled, until in the temperature of the atmosphere it was filled to within half an ounce measure of water. [In the continuance of this experiment, I was assisted by Dr. Black. In Dr. Robison's edition of Dr. Black's lectures, vol. i, page 147, the latter hints at some experiments upon this subject as made by him; but I have no knowledge of any except those which I made myself.]

"In repetition of this experiment at a later date, I simplified the apparatus by omitting the tube, and laying the flask upon its side in the oven, partly closing its mouth by a cork having a notch in one side, and otherwise proceeding as has been mentioned. I do not consider these experiments as extremely accurate, the only scale-beam of a proper size which I had then at my command not being very sensible, and the bulk of the steam being liable to be influenced by the heat to which it is exposed, which, in the way described is not easily regulated or ascertained; but, from my experience in actual practice, I esteem the expansion to be rather more than I have computed.

"A boiler was constructed which showed, by inspection, the quantity of water evaporated in any given time, and thereby ascertained the quantity of steam used in every stroke by the engine, which I found to be several times the full of the cylinder. Astonished at the quantity of water required for the injection, and the great heat it had acquired from the small quantity of water in the form of steam which had been used in filling the cylinder, and thinking I had made some mistake, the following experiment was tried:—A glass tube was bent at right angles, one end was inserted horizontally into the spout of a tea kettle, and the other part was immersed perpendicularly in well-water contained in a cylindric glass vessel, and steam was made to pass through it until it ceased to be condensed, and the water in the glass vessel was become nearly boiling hot. The water in the glass vessel was then found to have gained an addition of about one-sixth part from the condensed steam. Consequently, water converted into steam can heat about 6 times its own weight of well-water to 212°, or till it can condense no more steam. Being struck with this remarkable

fact, and not understanding the reason of it, I mentioned it to my friend Dr. Black, who then explained to me his doctrine of latent heat, which he had taught for some time before this period (summer, 1764); but having myself been occupied with the pursuits of business, if I had heard it I had not attended to it, when I thus stumbled upon one of the material facts by which that beautiful theory is supported.

Latent Heat.

"On reflecting further, I perceived that in order to make the best use of steam, it was necessary, first, that the cylinder should be maintained always as hot as the steam which entered it; and, secondly, that when the steam was condensed, the water of which it was composed, and the injection itself, should be cooled down to 100°, or lower, where that was possible. The means of accomplishing these points did not immediately present themselves; but early in 1765 it occurred to me, that if a communication were opened between a cylinder containing steam, and another vessel which was exhausted of air and other fluids, the steam, as an elastic fluid, would immediately rush into the empty vessel, and continue to do so until it had established an equilibrium; and if that vessel were kept very cool by an injection or otherwise, more steam would continue to enter until the whole was condensed. But both the vessels being exhausted, or nearly so, how was the injection water, the air which would enter with it, and the condensed steam, to be got out?

"This I proposed in my own mind to perform in two ways. One was by adapting to the second vessel a pipe reaching downward more than 34 feet, by which the water would descend (a column of that length over-balancing the atmosphere), and by extracting the air by means of a pump. The second method was by employing a pump, or pumps, to extract both the air and the water, which would be applicable in all places, and essential in those cases where there was no well or pit.

Air Pump.

"This latter method was the one I then preferred, and is the only one I afterwards continued to use. In Newcomen's engine, the piston is kept tight by water, which would not be applicable in this new method; as, if any of it entered into a partially exhausted and hot cylinder it would boil and prevent the production of a vacuum, and would also cool the cylinder, by its evaporation during the descent of the piston.

"I proposed to remedy this defect by employing wax, tallow, or other grease, to lubricate and keep the piston tight. It next occurred to me that the mouth of the cylinder being open, the air which entered to act on the piston would cool the cylinder, and condense some steam on again filling it; I therefore proposed to put an air-tight cover upon the cylinder, with a hole and stuffing-box for the piston-rod to slide through, and to admit steam above the piston to act upon it instead of the atmosphere. [N.B.—The piston-rod sliding through a stuffing-box was new in steam engines; it was not necessary in Newcomen's engine, as the mouth of the cylinder was open, and the piston stem was square and very clumsy. The fitting the piston-rod to the piston by a cone was an after improvement of mine (about 1774.)] There still remained another source of the destruction of steam, the cooling of the cylinder by the external air, which would produce an internal condensation whenever steam entered it, and which would be repeated every stroke; this I proposed

Steam Jacket.

to remedy by an external cylinder containing steam, surrounded by another of wood, or of some other substance which would conduct heat slowly.

Separate Condenser.

"When once the idea of the separate condensation was started, all these improvements followed as corollaries in quick succession, so that in the course of one or two days, the invention was thus far complete in my mind, and I immediately set about an experiment to verify it practically. I took a large brass syringe, 1¾ inches diameter, and 10 inches long, made a cover and bottom to it of tin-plate, with a pipe to convey steam to both ends of the cylinder from the boiler; another pipe to convey steam from the upper end to the condenser (for, to save apparatus, I inverted the cylinder). I drilled a hole longitudinally through the axis of the stem of the piston, and fixed a valve at its lower end, to permit the water which was produced by the condensed steam on first filling the cylinder, to issue. The condenser used upon this occasion consisted of two pipes of thin tin-plate, ten or twelve inches long, and about one-sixth inch diameter, standing perpendicular, and communicating at top with a short horizontal pipe of large diameter, having an aperture on its upper side which was shut by a valve opening upwards. These pipes were joined at bottom to another perpendicular pipe of about an inch diameter, which served for the air and water-pump; and both the condensing pipes and the air-pumps were placed in a small cistern filled with cold water. [N.B.—This construction of the condenser was employed from knowing that heat penetrated thin plates of metal very quickly, and considering that if no injection was thrown into an exhausted vessel, there would be only the water of which the steam had been composed, and the air which entered with the steam, or through the leaks, to extract.]

"The steam pipe was adjusted to a small boiler. When steam was produced, it was admitted into the cylinder, and soon issued through the perforation of the rod, and at the valve of the condenser. When it was judged that the air was expelled, the steam cock was shut, and the air-pump piston-rod was drawn up, which leaving the small pipes of the condenser in a state of vacuum, the steam entered them and was condeused. The piston of the cylinder immediately rose and lifted a weight of about 18 lb., which was hung to the lower end of the piston-rod. The exhaustion-cock was shut, the steam was re-admitted into the cylinder, and the operation was repeated, the quantity of steam consumed, and the weights it could raise were observed, and, excepting the non-application of the steam-case and the external covering, the invention was complete in so far as regarded the savings of steam and fuel.

"A large model with an outer cylinder and wooden case was immediately constructed, and the experiments made with it served to verify the expectations I had formed, and to place the advantage of the invention beyond the reach of doubt. It was found convenient afterwards to change the pipe-condenser for an empty vessel, generally of a cylindrical form, into which an injection played, and in consequence of there being more water and air to extract, to enlarge the air-pump.

Surface Condenser.

"The change was made, because, in order to procure a surface sufficiently extensive to condense the steam of a large engine, the pipe-condenser would require to be very voluminous, and because the bad water with which engines

are frequently supplied would crust over the thin plates, and prevent their conveying the heat sufficiently quick. The cylinders were also placed with their mouths upwards, and furnished with a working-beam and other apparatus as was usual in the ancient engines; the inversion of the cylinder, or rather of the piston-rod, in the model, being only an expedient to try more easily the new invention, and being subject to many objections in large engines."

Watt's Patent. Watt took out his patent in 1769 embodying the above improvements. His principal claims are as follows:—

"First, that vessel in which the powers of steam are to be employed to work the engine, which is called the cylinder in common fire engines, and which I call the steam-vessel, must, during the whole time the engine is at work, be kept as hot as the steam that enters it; first, by enclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water nor any other substance colder than the steam to enter it or touch it during that time.

"Secondly, in engines that are to be worked wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam-vessels or cylinders, although occasionally communicating with them; these vessels I call condensers; and, whilst the engines are working, these condensers ought at least to be kept as cold as the air in the neighbourhood of the engines, by application of water or other cold bodies.

"Thirdly, whatever air or other elastic vapour is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam-vessels or condensers by means of pumps, wrought by the engines themselves, or otherwise.

"Fourthly, I intend in many cases to employ the expansive force of steam to press on the pistons, or whatever may be used instead of them, in the same manner in which the pressure of the atmosphere is now employed in common fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the air after it has done its office.

"The fifth claim is not essential it being for a rotary engine not proceeded with.

"Sixthly, I intend in some cases to apply a degree of cold not capable of reducing the steam to water, but of contracting it considerably, so that the engines shall be worked by the alternate expansion and contraction of the steam.

"Lastly, instead of using water to render the pistons and other parts of the engine air and steam tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver and other metals in their fluid state."

CHAPTER IV

The Watt Engine

The steam engine as now designed and constructed by Watt was used, like the Newcomen engine, for the purpose of drawing water from mines, and the earlier engines of Watt did not differ much in general appearance from those of Newcomen, i.e. there was the same wooden beam, with the pumps at one end and the steam cylinder at the other, the connexions being linked chains going over radially curved surfaces at each end of the beam. The use of the separate condenser was, however, a very marked improvement, as by this means the alternate heating and cooling of the steam cylinder was largely prevented; again, the use of what he called "an external cylinder containing steam," now known as the steam jacket, the covering of the cylinder "with wood or other non-conducting material," and putting a cover with a stuffing-box on the top of the steam cylinder were, for the low pressures then used, most important improvements. But the engine still remained single-acting, and the piston-rod was connected to the engine beam by a linked chain going round the curved end of a beam the same as Newcomen's.

Double-acting Engine. The next great step forward was to make the engine double-acting, i.e. to make the piston-rod to push up as well as pull down, and for a very perfect means of doing this we are again indebted to James Watt, viz. for what is known as the "parallel motion." How Watt evolved the idea of the motion we do not know, and it is looked upon as rather a complicated piece of mechanism, but the following diagram (Fig. 18), will make it perfectly easy to understand.

Here let a be one-half of the working beam, that to which the piston-rod is attached; let a' and a'' be the centre lines of beam in its upper and lower positions. Let b be a rod of equal length with a, and working on a fixed centre

c, *b′* and *b″* being the upper and lower position of *b*. Let the centre *c* be fixed as much lower than the centre line of *a* as at least one-half the stroke of the engine. Let the end of the working beam *a* and the end of the radius rod *b* be connected together by the distance pieces *d* (one on each side of beam), and let the piston-rod head work on a pin or gudgeon in the centre of the distance pieces *d* (and between them).

Now, as the lengths of *a* and *b* are equal, the arcs each will describe will be equal; it is evident, therefore, that in going upward the upper part of the distance pieces *d* will incline to the left, while the lower part will incline to the right to exactly the same amount and in the same time; that being

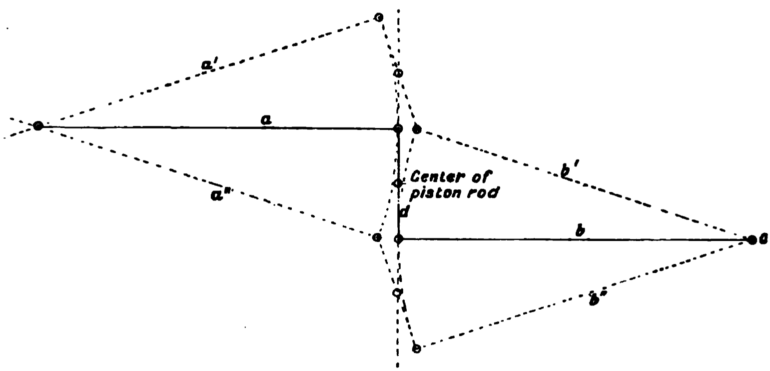


FIG. 18.

so, the centre, being equally distant from each end, will incline neither to the right nor to the left, but will travel in a straight line. The same thing occurs when the beam descends below the horizontal position. Again, the upper point of *d* inclines to the left, and the lower point inclines to the same extent to the right, the centre travelling in a straight line. The diagram (Fig. 18), shows clearly the three positions taken. The central position is shown in full lines, and the upper and lower positions in dotted lines.

Such an apparatus as Fig 18 for guiding a rod in a straight line was designed by the author for a machine he was constructing when but seventeen years of age, and it is so obvious, that it has doubtless occurred to many hundreds of engineers when seeking such a motion. For guiding the piston rod of a steam engine

it is, however, far inferior to Watt's motion, though mathematically more accurate. It has the disadvantage that the fixed centre of the radius rod *b* must project one-half the length of the working beam beyond the beam to the point *c*, and, apart from the greater length of engine house required on account of this, the fixed point *c*, on which the rod *b* works is somewhat difficult to obtain and retain. Watt's motion is free from these difficulties; drawn to the same scale it is shown in Fig. 19, and is one of the most interesting and beautiful pieces of mechanism in connexion with the steam engine. On referring to the diagram (Fig. 19), it will be seen that it contains the same elemental parts as those in

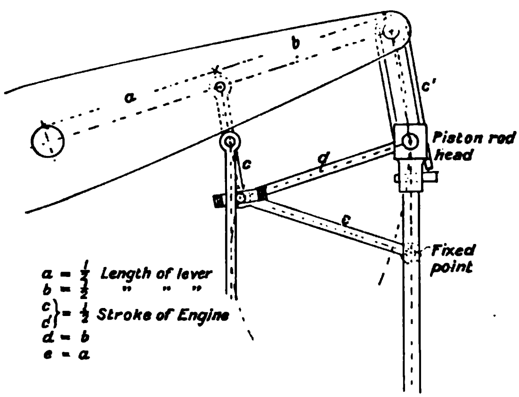


Fig. 19.

Fig. 18, but all is contained in about one-half the length. *a* which equals one-half the length of lever, corresponds to *a* in Fig. 18; *e*, the radius rod working on its fixed point, has a convenient attachment for this in the sides of the entablature which surrounds the working beam and corresponds to the radius rod *b* in Fig. 18. The centre of the length of the distance piece *c* is the true straight line, but the point of attachment of the piston-rod head moves in what is practically a straight line, the variation from straight being calculated to be less than $\frac{1}{1000}$ of the length of the stroke of engine when the motion is properly proportioned, but when this is done the beam may work within fairly wide angles. Fig. 20 shows a much wider angle of working, and it will be noticed that the distance pieces *c* and *c′* are much longer in this case than is

absolutely necessary if we would have a correct result. The distance pieces must always be at least half the length of the engine stroke, and in Fig. 20 the upper and lower positions, shown in dotted lines, show the straight line traversed by the lower end of *c′*.

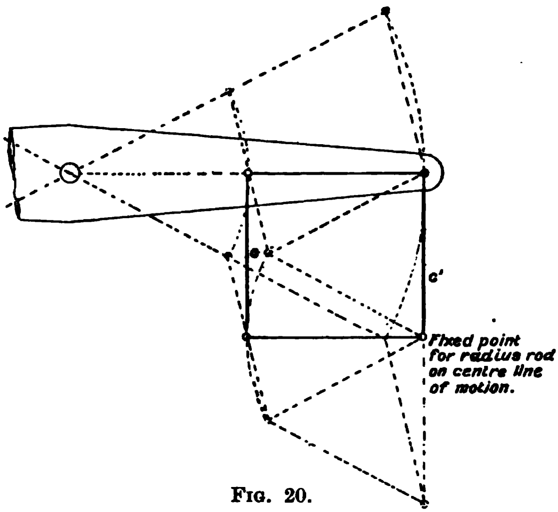


Fig. 20.

To show that this is essential we have in Fig. 21 shown the beam working at the same wide angle, but with short distance pieces, here it is seen that the upper point of the parallelogram at *a* is outside, and at its lower position at *a′* is inside the straight line, and further, that the radius rod cannot reach the end of distance piece at *c*. So that such a construction is not only inaccurate but impossible. We would say here that in graphic designs it is always wise to make an exaggerated diagram in order to discover and locate defects. Doubtless many of the supposed defects in Watt's parallel motion have been caused by incorrect proportion.

Fig. 22 shows the elemental principle same as Fig. 18, but proportioned for a very wide angle, at which it works equally accurately as at a smaller angle.

Up to this time the steam engine had been used principally, if not entirely, as an instrument for raising water out of mines, and, though of immense value, yet it was a somewhat cumbrous and wasteful engine for the purpose. Wasteful, i.e. in comparison with what was scientifically possible, but very eco-

nomical when compared with any other then known method of raising water. When, however, a method had been discovered by which the steam could be used on both sides of the piston, a new era began for the steam engine. With the same size of parts, it is evident that its power would be doubled, and if some means were adopted to turn the reciprocating motion of the beam into a rotary one, its use and utility would be indefinitely extended.

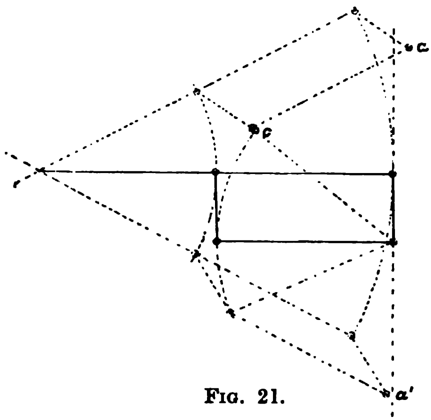


Fig. 21.

The Crank. The method of turning a reciprocating motion into a circular or rotary motion by means of a crank had been known for ages in the familiar foot lathe and scissors-grinding machines, and the use of the crank at once occurred to Watt, but he found to his annoyance that a patent for the use of cranks in connexion with a steam engine had been granted to one Peckard, and rather than make terms with the patentee, Watt devised what was known as the "sun and planet motion."

Sun and Planet Motion. In this apparatus, shown in Fig. 23, a toothed wheel *a* was keyed on to the shaft to be driven, and a similar toothed wheel *B* revolved on the crank-pin, but was secured to the connecting rod. The circular dotted line shows the path of crank-pin; the crank arm shown in dotted line behind the two pinions revolved freely on the main shaft, and carried the crank-pin *c* on its outer end. Being fixed to the connecting-rod, the wheel *B* is not free to revolve round the crank-pin, but the crank-pin in going round its circle revolves

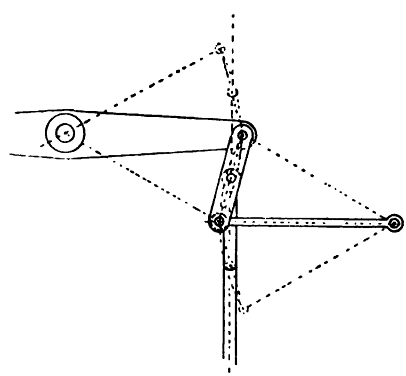


Fig. 22.

within the wheel *B* and serves principally to keep the two wheels properly in gear with each other, and also of course to regulate the length of stroke of the beam. On the connecting rod descending in the direction of the arrow, the wheel *a* is

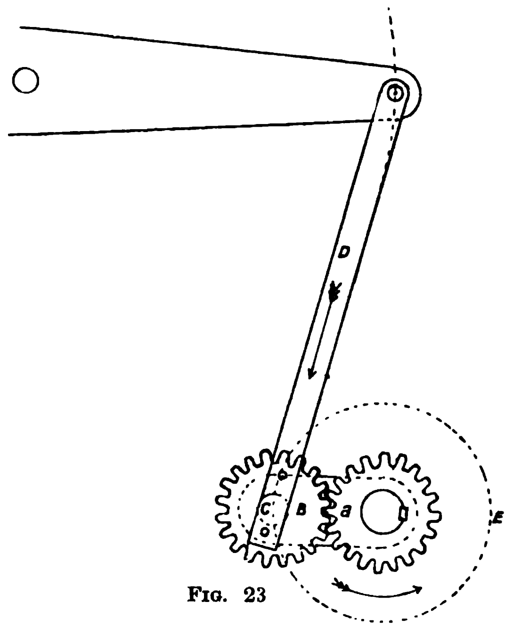


FIG. 23

compelled to turn round, due to the action of the teeth of *B*, and by the time *B* has gone round from the left to the right, i.e. to position *E*, *a* will have made one complete revolution and will make a second complete revolution by the time *B* has once more arrived at its starting-place, as shown in Fig. 23. Thus, for each revolution of the crank-pin there are two revolutions of the wheel *a*, and the shaft to which it is keyed. The foregoing is correct, of course, only if the relations of the toothed wheels to each other is as shown, i.e. with an equal number of teeth in each wheel. On the other hand if there were say 20 teeth on the crank pinion and 80 teeth on the wheel keyed to the shaft, then *a* and the shaft would make one revolution to each two of the crank ; while if *B* had 80 teeth and *a* had only 20, then *a* would revolve 8 times to one revolution of crank.

This power of increasing the speed of driving shafts was an incidental advantage when these engines were used for driving

quick-running machinery, as the engines made but few revolutions, 25 revolutions per minute being considered a fairly high speed. In other respects the apparatus was but a make-shift, and when the patent for the crank expired the latter was at once adopted by all engine makers.

Engine as Motive Power. The steam engine had now arrived at a state in which its use became economical for nearly every purpose for which power was required. Previous to this time the use of steam had been for the purpose of making a vacuum, the whole power of the engine resulting from the pressure of the atmosphere on one side of the piston, when condensation had caused a vacuum of the other; but now with mechanically operated valves, the parallel motion connexion between the piston-rod and the working beam, the cover to the upper end of the cylinder, the crank to convert reciprocating into rotary motion, and, further, the greatly improved methods of manufacture, the use of the direct pressure of steam naturally suggested itself and was rapidly adopted.

High and Low Pressure of Steam. Though Watt in his fourth claim (p. 37) showed that he was evidently alive to this use of high pressure steam, yet he seems to look upon it as a way to be adopted "in cases where cold water cannot be had in plenty," and he remained with several others a strong advocate of low pressure and condensation as the principal source of power. These two things went together for so many years that all condensing engines got to be called "low pressure," even though they might be using 50 lb. of steam, and all non-condensing engines were called "high pressure," even though the pressure of steam they were using was much less than this. As a matter of fact there is no necessary connexion between "low pressure" and condensing and "high pressure" and non-condensing, and at the present time there are many engines working at low pressures, i.e. under 40 lb., which are non-condensing, while the higher classes of condensing engines use pressures of from 80 to 200 lb. Engines should be described, therefore, as condensing or non-condensing without regard to the pressure of steam they work with.

Expansive Working of Steam. It is now time that we took under consideration the laws regulating the expansive use of steam. A solid weight, or a column of water, may be used to give a pressure; but steam is more than a pressure, it is

also like an elastic spring. We will first deal with it as a pressure only. Let us assume that we have a steam cylinder connected to an engine, and that the cylinder has a piston with an area of one square foot, i.e. 144 square inches, and a stroke of 4 feet, each stroke of this piston will correspond to the use of 4 cubic feet of steam and will absorb a certain number of heat units to produce it. If we want to have steam at a higher pressure we must use more heat and confine the steam

Relative Volume.

produced into a smaller space, or if we use the same space, then we must evaporate more water with which to make it. Thus, while one cubic inch can make one cubic foot at 212° of heat and atmospheric pressure, if we want a cubic foot of steam at 25 lb. pressure, we have to evaporate about 2 cubic inches of water, and its temperature will be 240°. If we want steam of 100 lb. we shall require 6 cubic inches of water and its temperature will be 327·5°.[1]

Power available from Relative Quantities.

Now let us see what power we can get from these various quantities of water? We will take first the steam at atmospheric pressure, i.e. 15 lb. Assuming that by means of our separate condenser we get a very good vacuum, leaving only 1 lb. absolute, then we shall have an effective pressure on our piston of 14 lb., and with a piston speed of 200 feet per minute we shall, by means of our standard formula, find the horse-power.

Area of piston	=144 square inches.
Pressure on piston	=14 lb.
Speed of piston	=200 feet per minute.

Then—

$$\frac{14\ (\text{lb.}) \times 144\ (\text{square inches}) \times 200\ (\text{feet per minute})}{33{,}000} = 12{\cdot}2 \text{ H.P.}$$

for 200 cubic inches of water converted into steam at a temperature of 212°.

We will compare this result with the other extreme high pressure, 100 lb. absolute. It will take six times as much water to make the same volume of steam as in the former case, we will see how much extra power we can get out of it.

[1] The above figures are only approximately correct as regards the relative volumes, but are sufficiently exact for illustration purposes, and it is more convenient to use simple round figures until the principles of expansion are clearly grasped. The exact relation between volumes pressures and temperature is given in the table in the appendix, p. 359.

Let the conditions be the same as before, then we shall have instead of 14 lb. $100-1=99$ lb. pressure, and $\frac{99\times 144\times 200}{33,000}$ $=86{\cdot}4$ H.P. with 1,200 cubic inches of water per minute, or 7 times as much power, for 6 times as much water, thus showing that, other things being equal, there is a slight gain in using high pressure steam; as, however, more fuel would have to be used in order to produce this high pressure the gain is more apparent than real.

If, however, instead of discharging this high pressure steam into the air or into a condenser we utilize it, the case becomes very different.

Expansion. Let us have the valve gear so arranged that when steam has been admitted for ⅛ of the stroke it shall be cut off and allowed to expand like the utilization of a spring exerting some pressure all the time, but the pressure becoming less as the piston nears the end of the stroke. Of one thing we are certain at once, viz. that we shall use only ⅛ of the volume of steam and therefore of water. We had 1,200 cubic inches before, now we shall have $\frac{1,200}{8}=150$ cubic inches only; we shall, of course, have less power, but it will not be reduced in the same proportion.

Boyle Law. According to the well-known Boyle law which was promulgated in 1662, "the volume of a portion of gas varies inversely as its pressure," i.e. the pressure of gas becomes smaller as its volume is increased, and the pressure becomes greater as the volume is decreased. Thus, let Fig. 24 represent the cylinder of one foot area and 4 feet stroke, we have shown it divides into 8 equal parts; into the first part of this we admit steam of 100 lb. pressure, and this part is shown shaded. According to the above law by the time the piston has advanced to the second division its pressure will have fallen to ½ or 50 lb., because its volume has doubled. We mark this position by a dot. Then by the time the piston has got to No. 4, the volume will have doubled again and the pressure fallen to another ½ or 25 lb. We again mark this position by a dot. We have now travelled half the total distance, and by the time the piston arrives at the end of the stroke, the pressure will have fallen another half or to 12½ lb. As it is a gradual fall, if we now count these points by a line we shall get a curve showing

the line of the fall. It will be noticed that at double the volume the pressure is half, at 4 times the volume it is quarter, and at 8 times the volume it is one-eighth; it therefore follows

Expansion Curve.

that at the intermediate numbers it will be $\frac{1}{3}$, $\frac{1}{5}$, $\frac{1}{6}$, and $\frac{1}{7}$ respectively. In Fig. 25 we have put in all these points and connect all by a line, the space diagonally hatched

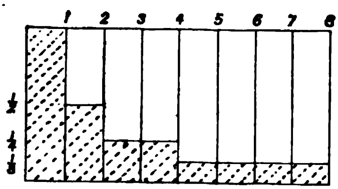


FIG. 24.

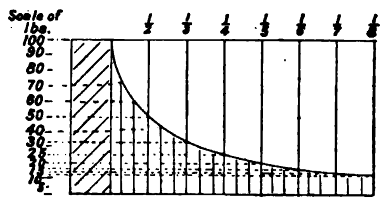


FIG. 25.

showing the steam used, and that marked by vertical lines the amount of utilization of the steam in expansion, the total shaded part showing the total pressure on the piston. We now want to know what is the average pressure on the piston, as we see it varies during the stroke. There are 4 ways in which we can find this out. We can take the middle of each space, and measure it, if we know the scale to which it is

Average Pressures.

drawn; we can consult some recognized table; or we can make our own calculations. It is best to learn all these ways. Having found the position of the dots and drawn our lines as in Fig. 25, we now divide each spaco

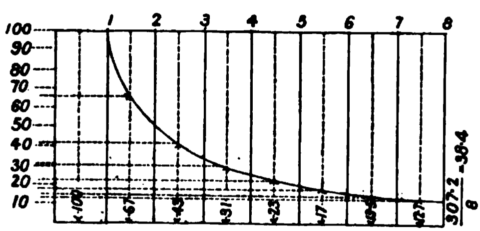


FIG. 26.

in half, as in Fig. 26, and measure the height along the centre lines of each division.

In the first division we have, of course, the full 100 lb., in the second 67, then 43, 31, 23, 17, 13·5 and 12·7, making a total of 307·2, which, divided by the number of divisions 8,

equals an average pressure of 38·4 lb. From this we have to take the same 1 lb. that we did in the former case, leaving a nett average pressure on the piston of 37·4 lb.

Tables. The second way we have indicated is to consult a set of tables, such as is given on p. 367 in appendix, from which we find at once that with 100 lb. absolute cut-off at ⅛ of the stroke, the average pressure on piston is 38·4 lb. Needless to say this second is by far the easiest and simplest, and is the one we shall follow in our future illustrations.

Geometrical Projection. A third way is by geometrical projection (Fig. 26A), A rectangular figure is drawn representing the cylinder—it may be of any proportions, but for our purpose preferably about

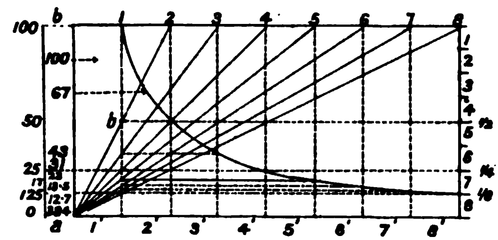


Fig. 26A.

twice as long as it is high—then the vertical line *a b* will represent the absolute pressure of steam within the cylinder, and it may be divided into as many parts as there are pounds in pressure. Let the figure be divided longitudinally into as many parts as equals the number of expansions as in Fig. 26A; this is 8. Then from each division let the vertical line 1, 2, 3, 4, 5, 6, 7, 8, be drawn. The enclosed space represents the 8 volumes of steam into which we have divided the cylinder. The first volume will be filled with steam of 100 lb. pressure, which by the time it has expanded as far as line 2, will have fallen to 50 lb. Now if we draw a line from 2 to the corner *a* of the figure it will cut line 1 exactly in half at *b*. From the intersection of this diagonal line with line 1 if a horizontal line be drawn it will cut line 2 in half; let a dot be made at that point. Then if from line 4 we draw a diagonal line to *a* it will cut line 1 at a point ¼ from the bottom, and if, therefore, a horizontal line be drawn from this intersection as shown, it will cut line 4 at ¼ of its length from the bottom. If from the extreme

end at 8 we draw a diagonal, it will cut line 7 at $\frac{1}{8}$ from the bottom, and a horizontal line drawn from it will cut line 8 at $\frac{1}{8}$ from the bottom, and so with all the others; let these be all drawn in, then we shall have a number of points, $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{4}$, $\frac{1}{5}$, $\frac{1}{6}$, $\frac{1}{7}$, and $\frac{1}{8}$ from the bottom of the figure, and if these dots be connected by a line we shall have formed what is known as a hyperbolic curve. This will exactly correspond with the curve in Fig. 26. If we now make a point on this curve exactly on the centre of each division and from this point draw horizontal lines to our scale we shall find the pressure at each of these points. The pressure, as so found, will be 67, 43, 31, 23, 17, 13·5, 12·7, in addition to the 100 lb., in first division, and the sum will be 38·4 lb. This exactly corresponds with what we got in Fig. 26. The advantage the geometric method has over that shown in Fig. 26 is that not only even, but any odd numbers of expansion can be taken, such as 11, 15, 17, 35, or others, and the curve is equally exact.

Hyperbolic Curve.

Mathematical Calculation.

The fourth way is by mathematical calculation and is far from simple, and considering the shape and nature of the curve on the card in Fig. 26 it is evident that no simple calculation can give the answer desired.

First the hyperbolic logarithm of the number of expansion must be found by an intricate calculation, to this must be added the No. 1, and the result is a constant.[1] Then divide the initial pressure of the steam used by the number of expansions, multiply this by the constant, and the result will be the average pressure.

To make this quite clear we will take the case we have been dealing with, 100 lb. of steam and 8 expansion, $\frac{100}{8}=12{\cdot}5$. The constant of 8 is 3·07, and $12{\cdot}5\times 3{\cdot}07=38{\cdot}375$, or as it is in the table, to the nearest hundredth in one place of decimals, 38·4.

The power we shall get out of this amount of steam will evidently be less than when we introduced it for the full length of the stroke, but we shall find the consumption of water is reduced in a much larger ratio.

Instead of an average pressure of 99 lb. we have now only

[1] A table of Constants is given in the Appendix, p. 373.

38·4, but our consumption of water is reduced to $\frac{1}{8}$ of what it was before; $\frac{1}{8}$ of 1,200 cubic inches is 150 cubic inches, thus we have the calculation $\frac{38{\cdot}4 \times 144 \times 200}{33{,}000} = 33{\cdot}4$ H.P. Thus

Relative Economy of Low and High Pressure Steam.

low pressure steam required 200 cubic inches of water to give 12·2 H.P., high pressure steam without expansion required 1,200 cubic inches of water to give 86·4 H.P., and high pressure steam expanded 8 times required 150 cubic inches of water to give 33·4 H.P. If we divide the power developed into the water used we find—

that $\frac{200}{12{\cdot}2} = 16{\cdot}39$ inches per H.P. per minute.

that $\frac{1{,}200}{86{\cdot}4} = 14{\cdot}8$ „ „ „ „ „

and that $\frac{150}{33{\cdot}4} = 4{\cdot}4$ „ „ „ „ „ [1]

So that, other things being equal, we find expanding steam 8 times is 3·36 more economical than using the same pressure of steam without expansion.

In the above illustration we have assumed a condenser which is essential to the working of very low pressure steam, but there are many cases where, from absence of water or other causes a condenser cannot be used, nevertheless the comparison will still be markedly in favour of the expansive use of steam. We must now use steam of over 15 lb. absolute pressure, and in order to make the comparison simple we will take 100 lb. in each case.

Steam used without Condensing.

One cylinder *A* shall have a piston area of one square foot, 144 inches, and cut-off at $\frac{1}{8}$ of the stroke; the other *B* shall have an area of $\frac{1}{8}$ of a square foot, and admit during the whole stroke, both having the same length of stroke, i.e. 4 feet. The volume of steam used per stroke must be the same for both of them, i.e. ·5 cubic feet of steam, for in the one case we have 4 cubic feet divided by 8=·5, and in the other $\frac{1}{8}$ of a square foot multiplied by 4=·5. With the large cylinder the conditions are the same as we considered in connexion with Fig. 26, and there we found that our average pressure on the piston was 38·4 lb.; from this we shall have to deduct not 1 lb.

[1] 27·73 cubic inches of water =1 lb.; 277·3 cubic inches =1 gallon.

as before, but 15, that being now our back pressure due to the atmosphere, leaving us a nett 23·4 lb. Our calculation, therefore, will be $\frac{23{\cdot}4 \times 144 \times 200}{33{,}000} = 20{\cdot}4$ H.P.

For the small cylinder we have $\frac{144}{8} = 18$ square inches area of piston, and 100 lb. $-$ 15 $=$ 85 lb. pressure on piston ; thus our calculation will be $\frac{85 \times 18 \times 200}{33{,}000} = 9{\cdot}2$ H.P.

In order to make our illustration complete we will now take the larger cylinder *C* with piston 144 square inches area, and work it with steam of $\frac{1}{8}$ the effective pressure that we did the small one, admitting during the whole length of the stroke. Now our effective pressure was 85 lb. and $\frac{1}{8}$ of this is 10·62 lb. This will, of course, give us exactly the same power as we got from the small cylinder with the higher pressure, but the amount of water used to make the steam will be different. The calculation will be $\frac{10{\cdot}62 \times 144 \times 200}{33{,}000} = 9{\cdot}2$ H.P.

We have now considered three ways in which steam may be used in a non-condensing engine having a stroke of 4 feet and the piston moving at the rate of 200 feet per minute, thus making 25 double strokes or revolutions per minute.

In one case, which we call *A*, we got 20·4 H.P. from ·5 cubic feet of steam per stroke $=$ 25 cubic feet of steam at 100 lb. pressure absolute working expansively.

In the second case, *B*, we got 9·2 H.P. from the same amount of steam and same pressure working without expansion.

In the third case, *C*, we had a cylinder 1 square foot area working with steam of 10·62 lb. pressure and without expansion.

We want now to ascertain what amount of water has been used in order to make these various volumes of steam. It must be remembered that the amount of water required to form steam varies with the temperature and pressure.

Specific Volume. In the table, p. 360, we see that the "specific volume," i.e. the number of cubic inches of steam produced from 1 cubic inch of water, is for steam of 100 lb. pressure 270, and for steam of 25·62 lb. 970 (the table gives 996 for 25 lb., the difference being the extra ·62 lb).

Thus in Case *A*—

20·4 H.P. is obtained from 25 cubic feet per minute or 1,500 cubic feet per hour of steam at 100 lb.

In Case *B*—

9·2 H.P. is obtained from the same amount of steam at same pressure.

In Case *C*—

9·2 H.P. is obtained from 200 cubic feet per minute or 12,000 cubic feet per hour of steam at 25·62 lb.

Dividing the total quantity of steam used per hour by its specific volume, we get for *A* and *B* $\frac{1,500}{270} = 5{\cdot}55$ cubic feet of water, and for *C* $\frac{12,000}{970} = 12{\cdot}37$ cubic feet of water.

There is a total lack of reason and correlation between our English weights and measures, so we find that a cubic foot of water weighs 62·3 lb. Then as *A* and *B* have used 5·55 cubic feet of water per hour $5{\cdot}55 \times 62{\cdot}3 = 342$ lb. and *C* has used 12·37 cubic feet $12{\cdot}37 \times 62{\cdot}3 = 770$ lb.

Thus *A*, using high pressure steam expansively, gives 20·4 H.P. for 342 lb. of water per hour, or 16·76 lb. per H.P. per hour.

B using high pressure steam without expansion, gives 9·2 H.P. for 342 lb. of water per hour, or 37·17 lb. per H.P. per hour.

C, using low pressure steam without expansion, gives 9·2 H.P. for 770 lb. of water per hour, or 83·6 lb. per H.P. per hour.

From the foregoing illustrations it is definitely shown that where there is no condensation, high pressure steam is more economical in use than low pressure, whether used expansively or not; and second, that it is much more economical to use high pressure steam expansively than not to do so.

The results we have given are, of course, theoretical only and are difficult if not impossible to be obtained in regular practice, but they are relatively correct; i.e. though a larger amount of water than is given above would be required to develop the given powers, yet they would all be increased in the proportion to the amounts, so that the absolute difference in favour of high pressure steam used expansively would be still great.

Steam in being used to do work is exposed to certain losses. Cast iron, which is the material almost universally adopted for

the construction of steam engine cylinders, is an exceedingly good conductor of heat. The ideal cylinder would be made of a material which was an absolute non-conductor, but this we cannot as yet obtain.

Thus when steam enters a cold cylinder, the latter rapidly abstracts heat from the steam until it has become as hot as the steam, and in so doing cools the steam and condenses a portion of it into water again; further, as the steam expands and does work, it cools, and the cylinder with it. The cylinder in cooling during the expansion of steam tends to raise the temperature of the steam again by imparting to it some of the heat it had got from it earlier, but then comes the time of exhaust. During the whole time of the exhaust (i.e. one-half of the time the engine is at work) the cylinder is in communication with the condenser or the atmosphere, and, of course, tends to cool down to that temperature, and thus has become cooler again before the next charge of hot steam is admitted into it. This process is repeated revolution by revolution of the engine.

So long as we use conducting materials, and so long as we use steam expansively, so long we shall have this loss; the question then is how to reduce it to its smallest dimensions.

One thing we can do and that is to surround the cylinder on the outside with some non-conducting material, so as to prevent the passage of heat outwards. This was formerly done, and is now, in many cases, by covering the cylinder with hair felt and wood strips or lags, hence it is called lagging the cylinder; and so long as only moderately high pressures were used, this was fairly satisfactory, but with the high pressures now in use, i.e. 200 lb. and over, such materials become useless, as they rapidly char and crumble away.

The temperature of steam of 100 lb. pressure is 327·5°, while that of 200 lb. is 381·6°, and wood and felt, though good non-conductors, are useless for such temperatures. Mixtures, therefore, of clay and fibre of various kinds in order to toughen it are used, being put on in a plastic state and covered when dry either with canvas, wood lags, or iron or steel sheets. If hair felt and wood lagging are used over the clay non-conducting material they give a perfect result, the conduction of heat outwards being so far prevented that not enough heat will pass even to melt a piece of butter placed on the outside.

The other loss, that due to internal condensation, may be minimised in another way, viz. that proposed by Watt, by surrounding the cylinder with another cylinder of metal, leaving a space between them, and filling this space with steam of the same temperature as that used in the cylinder. Fig. 27 shows such a jacket, *a* being the steam space between the two cylinders, *b* the outer casing of metal, *c* the earthy composition surrounding the casing, *d* the wood lagging, and *e* the thin outer covering of metal, which is put on more for ornament than use.

Steam Jacket.

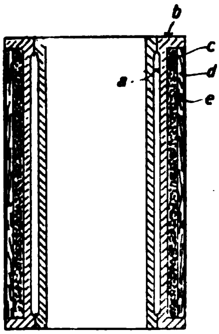


Fig. 27.

Lagging.

If properly lagged as shown, this steam jacket will not only prevent the radiation of heat outwards, but it actually gives out heat to the steam inside the cylinder whenever the internal temperature falls below that of the steam in the jacket. The first object is altogether good and economical; the second is good, but it is very doubtful if it is economical, and there are very conflicting opinions about it even amongst high authorities. One thing is certain, viz. that all the heat which is communicated from the jacket to the interior of the cylinder, is taken from the steam within the jacket and must be accounted for. One reason why there are such differences of opinion about the economy of steam jackets is because of the fact that an engine may, when working with a certain load and pressure, be more economical with steam in the jackets than without, yet the same engine with a different load and pressure is more economical without the jackets than with them, hence conflicting results of tests from incomplete experiments.[1]

The conclusion arrived at from the experiments of the author, is that a condensing engine working with comparatively low pressure and at a low duty, gains in economy from the use of the jacket, while when working with high pressures and full loads, there is a slight loss in economy from such use. In superintending official trials for economy with compound and

[1] See paper by Professor Dr. Mellanby, *Proceedings Institute of Civil Engineers*, vol. cxxxi.; also paper by Professor Calendar, *Proceedings Institution of Mechanical Engineers*, 1907, 2.

triple expansion condensing engines it is usual and advisable to have a preliminary, before an official test, and in all cases which have come under the author's notice, the full load trials are more economical in steam when the steam is not admitted into the jackets. The gain or loss, however, is but small, and with steam in the jackets of condensing engines all or nearly all cylinder condensation is avoided, and the engines work more smoothly and easily when the jackets are used, hence the statement made previously that "a steam jacket was good and sometimes economical." We shall refer to this subject later, when dealing with "Economy of Steam Consumption."

With regard to efficient lagging there is not and cannot be any question, everything is in its favour and its importance can scarcely be overestimated.

Surface Losses. The writer was once called in to investigate a case where a waterworks' pumping engine had been put up by a local engineer, and though according to all calculations the boiler was amply sufficient, yet as a matter of fact it could not supply enough steam to keep the pumping engine at work. The contractors were asked to put in a larger boiler, which they refused to do. In making the investigation the first thing done was to test the boiler's evaporation, in order to see if it could supply the quantity of steam specified to be required by the engine, and the tests showed that it could and did do much more than this. By mere calculation of volumes the engine cylinders should require much less than the amount, but they really required more. A very slight examination showed where the wrong was. The engine cylinders were neither jacketed nor lagged, nor was there any covering upon the long range of pipes connecting the engine with the boiler. The steam cylinder was directly connected with the pump, and thus the pistons worked at a slow speed. The condensation of steam from all these causes was so great that the cocks for discharging water of condensation from the cylinders had to be kept almost constantly open.

The users of the plant wanted the contractor to supply another or a larger boiler, but this was not at all necessary. As soon as these great losses due to radiation and condensation were remedied by properly clothing all exposed surfaces, the one boiler was found to be of ample capacity and power, and a very considerable saving was effected in the coal bill.

Condensation by Radiation a Factor of Time.

One of the reasons why the radiation losses were so great in this case was the very slow speed at which the pistons worked; these were connected direct to the plunger pumps and worked at a speed varying from 80 to 100 feet per minute, whereas the usual speed of steam engine pistons is now from 300 to 800 feet per minute. In connexion with this illustration let it be carefully noted that the loss due to radiation is largely a factor of the *time* during which the surface is exposed; thus if high temperature steam be admitted into a cylinder or pipe which is unlagged it will go on radiating heat from its surface at a rate depending upon the difference in temperatures during the whole time it is so exposed, whether any work be done within the cylinder or not. That being so it is evident that the more work that can be got out of such a cylinder within the same time, the radiation remaining constant, the less must be the relative loss due to radiation.

In order to make this matter quite clear let us take the case of two engines exactly alike in all points; let one work with a piston speed of 100 feet per minute develop 10 H.P. and use 250 lb. of steam per hour in doing its real work and waste 750 lb. per hour in radiation and condensation, which is quite possible and not altogether unknown. The losses from this cause are relatively enormous and we should have a nett consumption of no less than 100 lb. of steam per H.P. per hour.

With the other engine let the piston move faster, say at the rate of 500 feet per minute, and, other things being equal, this being five times faster than in the first case, we should develop five times as much power, or 50 H.P. instead of 10 H.P., that would take five times as much of the steam as was usefully expended, i.e. $5 \times 250 = 1,250$ lb. instead of 250 lb., but the steam lost would be no more than the 750 lb. lost in the former case, or a total of 2,000 lb., which divided by 50 H.P.=40 lb. per H.P. per hour, a not very wasteful consumption of steam in a simple high pressure engine. Now let us increase the speed to a 1,000 feet per minute, keeping all else the same, this will evidently double the effective H.P. and double the amount of steam used to do work, raising it from 250 to 2,500; but the loss due to radiation and condensation will be no more than at first, viz. 750 lb., making a total of $2,500 + 750 = 3,250$ lb., which divided by the 100 H.P. we should now have gives us

32·5 lb. of steam per H.P. per hour, a good and quite possible result from a simple non-condensing engine.

We stated above that the losses due to radiation and condensation should be "no more" at the high speed of piston than at the low speed, in point of fact the losses would be less. As we stated on p. 56, these losses are "a factor of the time and the difference of temperature." With a slow moving piston these differences of temperature are very marked, the piston moving slowly, the steam has time to heat up the cylinder to its own high temperature, losing its own heat in so doing. Again, during the exhaust stroke the cylinder has time in which to cool down to the temperature of the atmosphere, thus losing the heat given to it by the steam. Thus, the extreme ranges of temperature may be from steam of 100 lb. pressure= 338°, and that of the atmosphere say 60°, a difference of no less than 278°. It is true that in practice the cylinder never gets quite so hot as 338° nor quite so cold as 60°, but whatever the drop may be, it must be evident that the longer time it is exposed to these differences, the greater will be the drop. What we want here is to establish the very important fact, that, other things being equal, the more rapidly we can make the piston of a steam engine move, the more economical it will be in consumption of steam.

High Expansion. Hitherto we have dealt with no more than 8 expansions; we will now take a case of 16 expansions in a cylinder of 144 inches area and 4 feet stroke, and with a piston speed of 600 feet per minute, and we shall find that the gain due to expansive working is still greater.

The quantity of steam we shall use per stroke is $\frac{1}{16}$ of 4 feet, or $\frac{1}{4}$ of a cubic foot per minute; at 600 feet per minute and 4 feet stroke, we make 150 strokes per minute or 75 revolutions. Then $150 \times \cdot 25 = 37\cdot 5$ cubic feet of steam per minute or 2,150 cubic feet per hour.

The specific volume of steam at 215 lb. absolute is 125·6, so $\frac{2,150}{125\cdot 6} = 17\cdot 1$ cubic feet of water, which at 62·3 lb. per foot= 1,065·33 lb. of water used per hour.

The power we get out of this engine according to our formula will be as follows :—

Steam pressure, 215 absolute.

Piston speed, 600 feet per minute.

Cut-off at $\frac{1}{16}$ of the stroke.
Constant of 16=3·76.

Then $\frac{215}{16} \times 3{\cdot}76 = 49{\cdot}49 - 1 = 48{\cdot}49$ average effective pressure

and $\frac{48{\cdot}49 \times 144 \times 600}{33{,}000} = 126{\cdot}9$ H.P.

As we found that we used 1,065·33 lb. of water per hour this amount divided by the horse-power will give us the pounds of water used per H.P. and $\frac{1{,}065{\cdot}33}{126{\cdot}9} = 8{\cdot}4$ lb.

This is a much smaller quantity than is found requisite in practice to give one horse-power by expanding steam in one cylinder; in our next chapter we will try and find the cause, and also the means by which we may most nearly approach to this theoretical possibility.

CHAPTER V

The use of Steam in Multiple Cylinder, or Compound Engines.

HAVING now satisfied ourselves of the great advantages of "high pressure of steam," "high speed of piston," and "great expansion," we will now consider how these advantages can be obtained in the best way, i.e. without corresponding disadvantages.

Steam Engine Economy. We saw in our last chapter that by using steam of 200 lb. pressure and expanding it 16 times, we could theoretically get 1 H.P. from the use of only 8·4 lb. of water or steam per hour, but this, we stated, was an economy impossible to be obtained in practice. Even with the most careful jacketing, and the most perfect lagging, an engine working with 16 expansions in a single cylinder uses about 3 times this weight of steam; it follows, therefore, that from some cause only about 30 per cent. of the steam is used, and 70 per cent. of it is wasted in such an engine, and a very large proportion of this waste is due to radiation, conduction and condensation losses.

Cylinder Temperatures. We have already seen how, when we expand steam many times, that the temperature within the engine cylinder drops very considerably. This drop in temperature is not only absolutely greater when the difference is great, but it is also relatively greater; a reference to the table in the Appendix, p. 360, will show that while the difference in temperature between 200 lb. and 100 lb. pressure is that between 381·6° and 327·5°, or only 54·1°, the difference between 100 lb. and 1 lb. pressure is that between 327·5° and 102·1°, or no less than 225·4°. The higher we go in pressure the smaller is the difference in temperature between each pound.

At this point it would be well to examine the table in question, which has been compiled with great care, and from which it

will be seen that the rise in temperature from the zero of no pressure to that of 1 lb. is 102·1°, while from 1 lb. to 2 lb. it is 24·2°. When we reach 100 lb. pressure, the rise in temperature for the next pound is only ·7 of a degree, from 200 lb. to 201 lb. it is ·4°, from 300 lb. to 301 lb. it is ·3°, from 400 lb. to 401 lb. only ·2°, and when we reach as high as 1,000 lb., the rise is slightly under ·1°.

It follows, therefore, that the less difference we can have between the temperature of the entering steam and that of the exhaust, the smaller will be the losses due to condensation and radiation.

Multiple Cylinder. It is obvious that we can do this by utilizing the high pressure steam first in a cylinder when it shall be expanded only 4 times instead of 16 times, and then take the exhaust and utilize it in another cylinder, expanding it again 4 times. By so doing we shall have got the same expansion out of it, i.e. $4 \times 4 = 16$, as if we had expanded it in one cylinder, but we shall not have exposed it to the same differences in temperature. Assuming that we start with steam at 200 lb. pressure absolute by expanding it 16 times, we shall have reduced its pressure to $\frac{1}{16}$ of 200 lb. or 12·5 lb. In the first cylinder we shall have reduced it from 200 lb. to 50 lb., and in the second from 50 lb. to 12·5 lb. In the first cylinder we have a total drop of 100·2°, and in the second cylinder a drop of 77°; of course the total drop in temperature is the same, viz. 177·2°, but as the *rapidity* of the fall is so much greater when the difference is great, and we have reduced this difference to practically half in the same time, we have secured a great advantage. Assuming the rapidity of the fall to be as the square of the difference in temperature (and it is at least that), then in the one case we have the square of 177·2, i.e. 31,400, as against the sum of the squares of 100·2 and 77, i.e. 10,040 + 6,930 = 16,970, or a little more than one half. If we go further and use three cylinders, following the same method of calculation, we find that we have reduced the loss to about $\frac{1}{3}$ of that when one cylinder only is used, other things being equal—but other things are not equal. If we expand steam 16 times in one cylinder, we are doing cooling work during the whole expansion, which is $\frac{15}{16}$ of the whole stroke, when if we do it in two we are admitting steam during $\frac{4}{16}$ of the stroke and only expanding during $\frac{12}{16}$; thus we have two savings, one due to the smaller

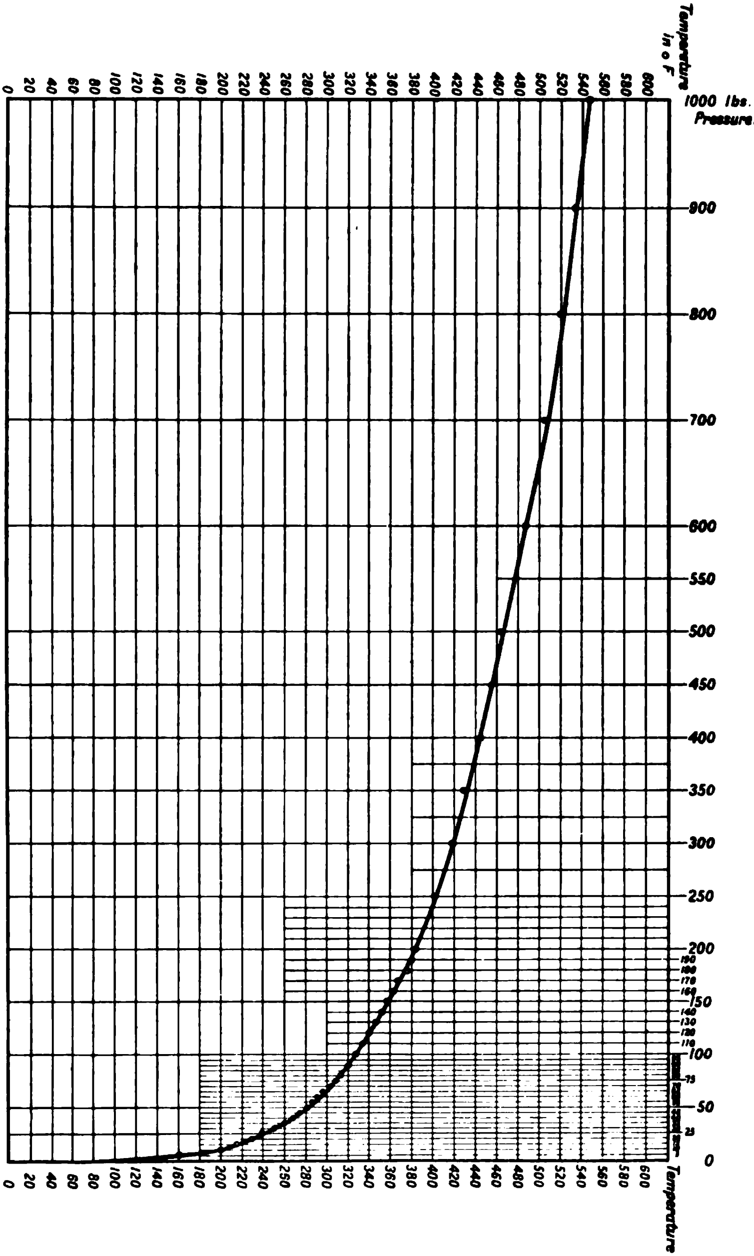


Diagram of Pressures and Corresponding Temperature.
Reduced from Appendix.

differences in temperature, and the other due to the shorter time in which the expansion takes place.

Compound Engine. The result of these reductions in losses is shown by the much greater economy in working of compound or multi-cylinder engines. When only two cylinders are used, such engines are called "compound." When three are used they are called triple-expansion, and when four are used they are called quadruple-expansion.

Triple Expansion.

When the proportions are correct and the design of a compound condensing engine is properly carried out, it is found in practice that they can produce a horse-power from about 16 lb. of steam per hour. This corresponds to a reduction of the losses by about one half.

An even better result is sometimes obtained: the quantity varies with the pressure of steam and the size of the engine; other things being equal, the larger the engine, the smaller will be the consumption of steam, because the smaller are the losses due to conduction and radiation. This is due to the fact that the area of a cylinder and therefore the effective power of an engine, increases according to the square of its diameter, while its circumference and corresponding surface, increases only directly as its diameter. Thus the area of 1 foot diameter is ·7854, and the area of 2 feet is 3·1416 or four times as much, while the circumference of 1 foot is 3·1416, and the circumference of 2 feet is 6·2832 or only twice as much.

According then to its size and the pressure of steam, the compound condensing engine will require from 15 lb. to 18 lb. of steam per H.P. per hour, and the triple-expansion engine from 10 lb. to 14 lb.

There are some losses due to clearances, wire drawing, and other causes which cannot be quite eliminated, but the practical result of 10 lb. obtained with a triple-expansion engine nearly approaches the 7·0 calculated.

Equalization of Strain. Another advantage to be obtained by expanding steam in two or more cylinders instead of one, is a mechanical one, viz. that of reducing the strains on the working parts; this we shall deal with later. We will now take as illustration an engine working with steam of 145 lb. above atmosphere, i.e. 160 absolute. We will first deal with the steam working in a high pressure, i.e. a non-condensing engine. In this case our "back pressure," i.e. the pressure on the exhaust

side of the piston, will be at least that of the atmosphere, say 15 lb., it is advisable that it should be at least 1 lb. more than that, so that it may have a tendency to rush out rather than to let the atmosphere rush into the cylinder, as it would be sure to do were the pressure less inside than outside.

If now we divide our total pressure of 160 lb. by 15 lb. (the atmospheric pressure, we shall get 10·2), which number of expansions, or even somewhat more, we might use were we exhausting into a vacuum; but as we discharge into the atmosphere it will not be advisable to use more than 9 expansions. Now 160 lb. ÷ 9 leaves 17·7 lb., or 2·7 lb. above the atmosphere. It may be asked here, why not expand more than this? since high expansion is good. Because it must not be forgotten that the atmosphere, thin, and almost impalpable as we seem to find it, does really exert the pressure of 15 lb. on each square inch of all surfaces with which it comes into contact. As we are entirely surrounded by it, i.e. pressed equally on all sides, and as it is within as well as without us, we are sensible of no pressure; but the experiment we tried in the second chapter and described on p. 23, shows how great is its power when unbalanced by a corresponding pressure on the other side of the surface exposed to it. Always, therefore, when designing or calculating the power of a non-condensing engine, we must take the pressure of the atmosphere into account, and exhaust at something above this point.

We are going to use the steam in two cylinders, and want to make the best use of it, for this reason we must exhaust from the high pressure cylinder at the same pressure that we want to use in the low pressure, or what is the same thing, we must use the steam in the low pressure cylinder at the same pressure at which it is exhausted from the high. This is so obvious at first that it seems unnecessary to state it; but it is not so simple as it seems, and it is seldom done so perfectly as it ought to be in order to secure the greatest economy in working. It is evident that our two cylinders must be of different sizes; that in which we use high pressure steam being small, and that for low pressure steam proportionately larger. There is by no means a uniform practice on the relation between the two cylinders, the reason being that there should be a definite proportion for each different pressure of steam.

Relative Sizes of Cylinders.

Rule for Ratio of Cylinders. A good practical rule is to find the number of expansions that can be wisely made with the pressure of steam that is available, and take the square root of this for the relations of the cylinders to each other.

Thus if we had a high enough pressure for 16 expansions, the square root of 16 being 4, we should make the areas of our cylinders in the relation of 4 to 1. In the case we are now dealing with we have decided upon 9 expansions, and the square root of 9 being 3 we will make our cylinder 3 to 1, by this means we shall in our first cylinder, expand the 160 lb. pressure 3 times, reducing it from 160 lb. to 53·3 lb., and this again expanded 3 times in the large cylinder will reduce it from 53·3 lb. to 17·7 lb. absolute; from this we deduct the atmosphere, 15 lb., leaving us 2·7 lb. as the pressure of our exhaust. It will be well now to look inside these two cylinders, and ascertain what pressure we get at various parts of the stroke.

Internal Pressures. We will assume a stroke of 36 inches, and a diameter of 13·6, which will give us an area of 1 square foot, or 144 square inches; let this be represented by Fig. 28. Up to the point of cut-off, i.e. $\frac{1}{3}$, the pressure will remain equal, i.e. 160 lb., from that point it will begin to fall according to the law given on p. 46. By the time then that the steam has filled double the space, i.e. up to 24 inches, it will have fallen to $\frac{1}{2}$ the pressure, or 80 lb., and equally it is evident that by the time it has reached the end of the stroke, and thus fills 3 times the space, it will have fallen to $\frac{1}{3}$ of the pressure, or 53·3 lb.

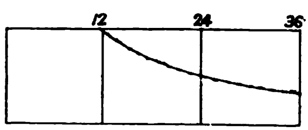


FIG. 28.

A little consideration of these facts will enable us to understand a simple formula by means of which the pressure can be obtained at any part of the stroke. Let the initial pressure be multiplied by the number of inches travelled by the piston up to the point of cut-off, and then divided by any other point or space required, the result will be the pressure at the point required. If then we let P stand for pressure, N for number of inches travelled to point of cut-off, S for space or number of inches at which the pressure is wanted, then the formula will be $\frac{P\ N}{S}$ = equal pressure at required point.

Testing our results by this formula we find they correspond. Thus $P=160$ and $\frac{160\times 12 \text{ inches}}{24}=80$, the pressure at 24 inches of the stroke, and $\frac{160\times 12}{36}=53{\cdot}3$ the pressure at the end of stroke as in Fig. 28.

Now we will try the intermediate spaces $160\times 12=1{,}920$, and this number divided by any number of inches will give us the pressure at that point.

The first 4 spaces are 160·0.		160·0
		160·0
		160·0
		160·0
The centre of the next space will be 13·5 inches, and 1,920÷13·5	=	142·2
1,920÷16·5	=	116·3
1,920÷19·5	=	98·4
1,920÷22·5	=	85·2
1,920÷25·5	=	75·2
1,920÷28·5	=	67·7
1,920÷31·5	=	60·9
1,920÷34·5	=	55·6

The total of these 12 pressures is 1,341·5, which divided by the number of spaces gives the average pressure, 111·7. Having now explained the method of obtaining any pressure at any point of expansion, we shall in future refer to our tables whenever they serve.

If these various pressures be plotted out to scale, and a dot put in each division to show the height as in Fig. 25, we shall find that when these dots are all united by a line that the line forms a curve which is known as an adiabatic line. This means a line of pressure which the expansions of the steam would give, if it neither received nor parted with any heat during the stroke. This line is never reached in practice, for the reasons given in the preceding pages, but in the best designed engines it is very nearly reached, and it will best serve us for all our illustrations. We will first plot

Adiabatic Curve.

Adiabatic Curve from two Greek words, "δια" and "βαινειν"

the curve as it would be if all the expansions took place in one cylinder, then deal with the two cylinders separately and afterwards combine them into one diagram.

Fig. 29 shows the action of 160 lb. of steam when expanded 9 times in one cylinder and cut off at $\frac{1}{9}$. The terminal pressure will then be $\frac{1}{9}$ of 160 lb. or 17·7, and at the other main divisions it will be $\frac{1}{8}$, $\frac{1}{7}$, $\frac{1}{6}$, $\frac{1}{5}$, $\frac{1}{4}$, $\frac{1}{3}$, and $\frac{1}{2}$ respectively; these points are set off graphically, the intermediate points indicated by the

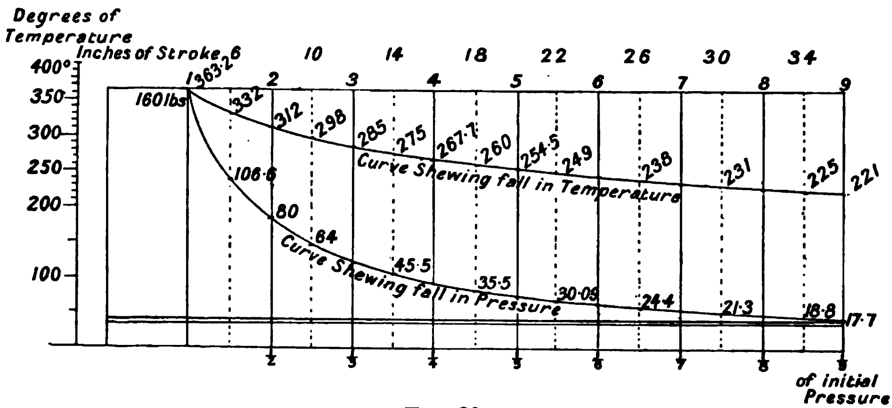


Fig. 29.

number of inches travelled, i.e. 6, 10, 14, 18, 22, 26, 30, and 34 are calculated according to the formula given on p. 48, and it will be seen that they all correspond and fit in with the required curve. The exhaust is shown on the atmosphere line, being 2·7 lb. below the terminal pressure.

At the left hand of the diagram in Fig. 29 is a scale of temperature up to 401°, and with the scale we have plotted off the temperatures at all the points in the steam pressure line. It will be noticed how much less rapidly the temperature falls than the pressure, but still there is a considerable drop from 363·2° to 221°, viz. 141·2°.

Temperature Losses. Fig. 30 shows what would take place if, instead of using the steam in one cylinder of 3 square feet area, cutting it off at $\frac{1}{9}$, we put our high pressure steam into a cylinder of 1 square foot area, and cut it off at $\frac{1}{3}$, thus letting it fill three of the spaces instead of one. The fall in temperature would then be from 363·2° to 255° only, or 108·2° instead of 141·2°.

If properly designed (as we shall show later) the exhaust

line will be practically a straight one, as shown in Fig. 30, and the discharged steam will be delivered at a pressure of 53·3 lb. This we call the high pressure, or H.P. diagram. There are

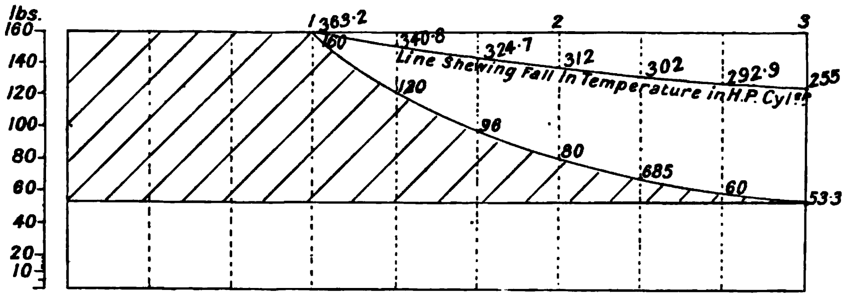


Fig. 30.

3 cubic feet of steam contained in a cylinder 1 foot in area and 3 feet stroke, so this quantity is just enough to fill a cylinder having 3 feet area to 1 foot of its stroke.

The diagram of the low pressure or L.P. cylinder, is shown in Fig. 31. Then starting with a pressure of 53·3, the steam is admitted to *a*, the point of cut-off at $\frac{1}{3}$ of the stroke, from which point it is expanded down to 17·7 lb., exactly as the same volume did in Fig. 29.

Power Found by Low Pressure Cylinder.

If we calculate the power we get from this diagram, we shall find that whether the steam is used in one cylinder, as in Fig. 29, or in two, as in Figs. 30 and 31, we get exactly the same power from 160 lb. of steam expanded 9 times.

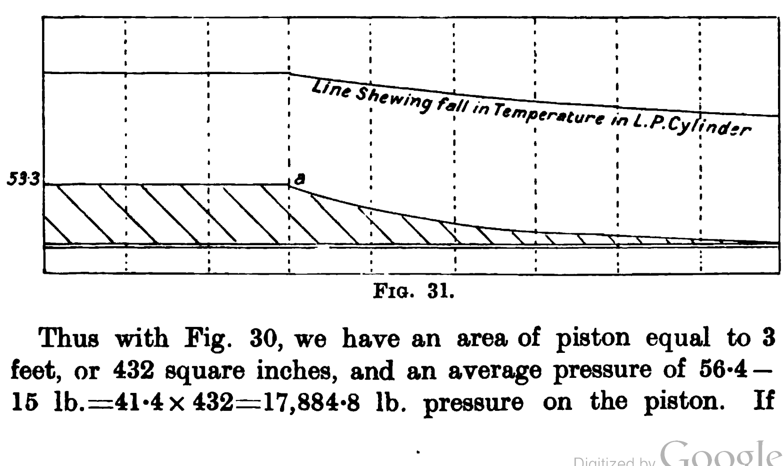


Fig. 31.

Thus with Fig. 30, we have an area of piston equal to 3 feet, or 432 square inches, and an average pressure of 56·4 − 15 lb. = 41·4 × 432 = 17,884·8 lb. pressure on the piston. If

we take the two cylinders we have in the H.P. an area of 1 foot, or 144 inches ; an average pressure of 111·5 − 53·3 = back pressure, or 58·2 lb., 144 × 58·2 = 838.08 lb. The L.P. has an area of 3 feet, or 432 inches, with an average pressure of 37 − 15 = 22. Then 432 × 22 = 950·4 as the pressure on the L.P. piston, and 950 + 838 = 1,788 total pressure on the pistons. Granted then equal speeds of pistons, the power will be exactly equal, and were both worked condensing there would be exactly the same additional power, i.e. 14 lb. extra, on an area of 432 inches in each case.

The only difference then is, as we have before pointed out, that, in the compound engine, we expose the steam to smaller differences in temperature.

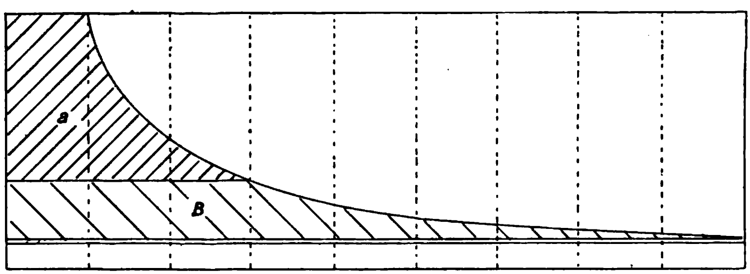


Fig. 32.

Combined Diagrams. Fig. 32 shows how the diagrams of the H.P. and L.P. cylinders can be combined. It will be noticed that the L.P. diagram remains exactly the same, but the H.P. shown in Fig. 30, being from a piston only 1 foot in area as against the 3 feet in the L.P. is reduced in length in the same proportion, the part *a* being the H.P. and the lightly shadowed part *B* being the L.P.

The three diagrams we have dealt with, show the behaviour of steam in compound engines when provision is made to maintain a steady uniform pressure for the L.P. cylinder, as can very easily be done in the case of cross compound engines, with cranks at right angles with each other; in tandem engines, the steam is frequently allowed to flow directly from the H.P. cylinder to the L.P., the piston moving in the same times. When this is done it is evident that the exhaust line of the H.P. must fall as the L.P. piston advances. Thus by the time the L.P. has reached the point of cut-off at *a* in Figs.

33 and 34, the steam is filling $\frac{1}{3}$ of the L.P. and also fills $\frac{2}{3}$ of the H.P., so that there will be a space of 5 cubic feet to contain 3 cubic feet of steam. That being so, its pressure

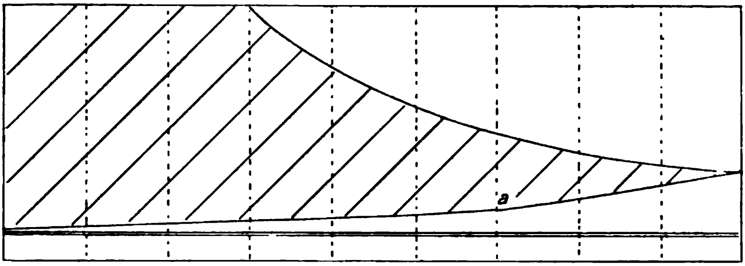
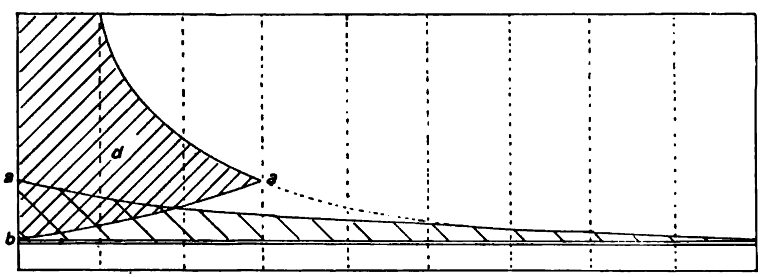


Fig. 33.

must fall to $\frac{3}{5}$ or about 32 lb. instead of 53·3, and from the point of cut-off it falls gradually to 17·7 lb. at the end. When these two diagrams are combined by dealing with them as we did with 30 and 31 in Fig. 32, their relative proportions are shown in Fig. 34, the H.P. being *d* and the L.P. *B*.

In this combination we have not avoided the same extremes of temperature as we did with Figs. 32 and 33, and though the total power is practically the same, the economy secured by such an arrangement of tandem compound is not so much as is obtained by the use of the steam, as shown in Figs. 31, 32, and 33.

In the above illustrations we have only dealt with non-condensing engines exhausting slightly above atmospheric



Tandem
Fig. 34.

pressure. As this pressure is practically 15 lb., if it can be removed by exhausting the steam into a vacuum, it is evident that we should have 15 lb. extra available pressure on the

piston of the low pressure cylinder, as any pressure removed from one side of it produces the same effect as an equal pressure added to the other side.

So important is the employment of a vacuum that, as we have shown in the earlier chapters, the only use that was made of steam by the first makers of steam engines was to drive out the air in the cylinders, filling it with steam instead, and then by the condensation of this steam, to produce a vacuum.

The Vacuum. As steam pressures have got higher and higher the need for a vacuum has become less and less, and by far the greater number of steam engines in use exhaust above the atmospheric pressure into the air. The proper use of a condenser for producing a vacuum is never-

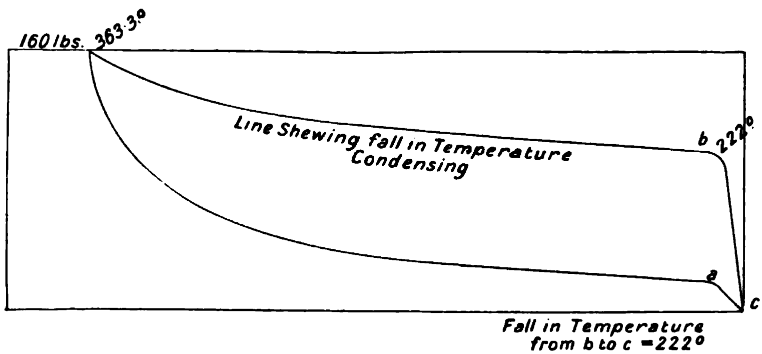


FIG. 35.

theless a considerable aid to economy, and, in the case of steam engines used for propelling ships, condensation is a prime necessity, not only for economical working, but for the sake of recovering in the form of fresh water the steam which has driven the engine, so that it may be used over again in making steam. Sea water being so strongly impregnated with salt is unsuitable for the purpose, owing to the large sediment of salt it would leave behind.

We cannot, however, have the advantage of a vacuum in a reciprocating engine without some disadvantage, such as the greater complication of its parts, the power required to drive the air pump and, above all, the much greater fall in temperature. Although by the use of the separate condensing chamber a great part of the loss is removed, yet not the whole

of it. During the whole of the exhaust stroke, which is half the time the engine is working, the low pressure cylinder is in direct communication with the condensing chamber through the exhaust pipe, and is being rapidly cooled by it. Were a perfect vacuum attained, the cooling effect would be much more severe than it is in practice ; Fig. 35 shows (to the same scale as Figs. 29 to 34) how great is the fall in temperature to a perfect vacuum. Thus during the whole of the steam stroke whilst the steam pressure falls from 160 lb. to 17·7 lb., the fall in temperature is only from 363·3 to 222 or 141·3 degrees. Yet when a condenser is used and a perfect vacuum obtained, the extra fall in temperature is 221° for only 17·7 more lb. in pressure, this is a very heavy price to pay for such a gain.

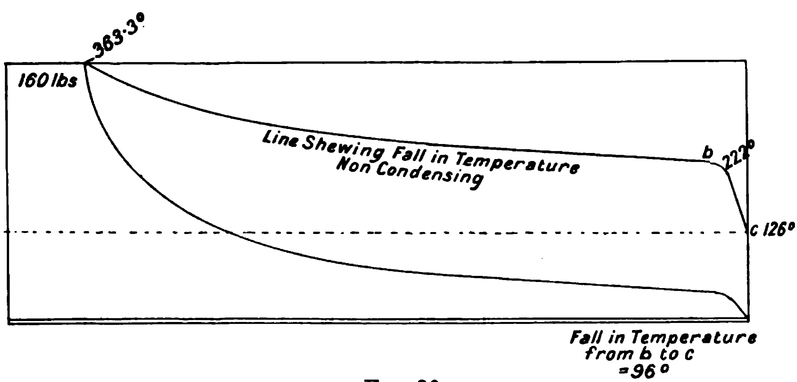


FIG. 36.

Rapidity of Temperature Drop. Not merely the extent but the rapidity and suddenness of the fall is clearly shown in Fig. 35, where *a* on the steam line shows the opening of the exhaust valve just before the end of the stroke, *b* shows the amount of the fall up to the opening of the communication with the condenser, and *c* shows the great and sudden fall from that point.

In practice a perfect vacuum is never obtained, but attempts are often made to get as near to it as possible—well-meaning but mistaken attempts. 13 instead of 15 lb. of vacuum is as much as makes for real economy, and Fig. 36, shows how much less is the fall in temperature when this moderate vacuum is worked with. We have only lost $\frac{2}{15}$ of

the extra pressure, but the fall in temperature is less than ½ what it was with the higher vacuum, being only 96° instead of 222°.

Reciprocating versus Continuous Action Engines. The above remarks apply only to reciprocating engines, when the cylinder is alternately in communication with the boiler and the condenser. One of the latest developments of steam engine manufacture is the turbine, the construction of which we shall deal with later on. In this engine the flow of steam is continuous from the highest pressure to the vacuum, each part being exposed to its own temperature only, and thus the higher the pressure at which it begins, and the lower that at which the steam leaves the engine, the more economical is the result. Though with the turbine there are several

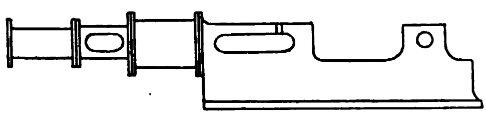


FIG. 37.

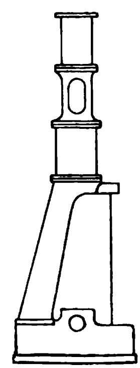


FIG. 37A.

sources of loss which are avoided in reciprocating engines, yet when starting with a sufficiently high pressure and exhausting into a good vacuum, the turbine will give results equal to those obtained with the best of the former.

Reduction in Strain by Compounding. Having now shown how compounding our engine increases our economy, we will examine how it reduces our strains. In our first illustration we will assume that the two cylinders are placed in a line with each other as in Figs. 37 and 37A, or "tandem," either vertically or horizontally, so that we shall have to deal with the sum of the strains on the two pistons. For purposes of comparison, 1 square inch and 3 square inches will serve as well as any higher numbers, so we will use these in

each comparison. In the high pressure cylinder then we have $(1 \times 160) - 53{\cdot}3$ the terminal pressure, or a nett pressure of 106·7 lb. In our second cylinder the pressure will be $53{\cdot}3 - 15$ for the atmosphere $= 38{\cdot}3 \times 3 = 114{\cdot}9$ lb., and thus the nett steam pressure on the two pistons at the beginning of the stroke will be high pressure $106{\cdot}7 + 114{\cdot}9$ for the low pressure, or a total of 221·6 lb.

Now if we had to expand the same pressure of steam down to the same terminal pressure in one cylinder, it would have to be of the same dimensions as the low pressure of the compound. The number of expansions being the same, viz. 9, we should cut off at $\frac{1}{9}$ of the stroke; it would have 3 square inches area, and $(160 \text{ lb.} - 15) \times 3$ inches would give us a nett 435 lb. pressure, practically double what the strain was with the two cylinders.

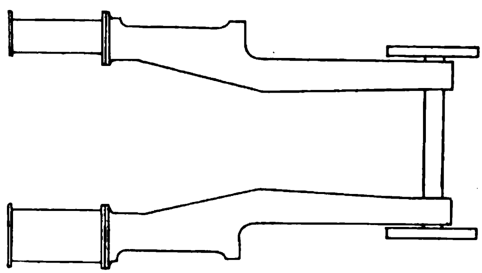


FIG. 37B.

When, however, we place the cylinders side by side, or cross-compound as they are called, like Fig. 37B, which is a much more usual practice, then the difference in strains is still greater. Instead of the sum of the two cylinders we now have to deal with only one, and then we find that the strain in a well-designed cross-compound engine are only about $\frac{1}{4}$ of what they would be in a single cylinder engine, using the same pressure of steam and expanding it the same number of times. This difference can be best realized by comparing the strains as they are shown graphically in the following diagrams, Fig. 38 showing the 435 lb., Fig. 38A showing the sum of the two cylinders in a tandem engine, and Fig. 38B showing the strains on each cylinder of a cross-compound.

So far we have dealt only with two-cylinder compound engines, and with steam of 160 lb. pressure there is not suffi-

cient extra economy to be gained by using more than two cylinders to pay for the additional complication. 160 lb. pressure is far below our limits of pressure and as further economy can be obtained by still higher pressures they are rapidly coming into use.

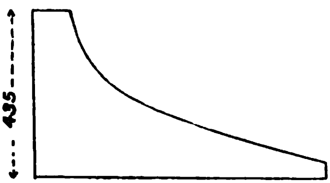


FIG. 38.

FIG. 38A.

FIG. 38B.

Effect of High Temperature on Lubricants.

As far back as the year 1871 the author built an engine to use steam of 315 lb. pressure, generated by a Perkins' tubular boiler with the fire outside the tubes; the engine was triple-expansion and expanded the steam 27 times, i.e. three times in each cylinder. Now steam of 315 lb. pressure means a temperature of 421°, little less than the melting point of tin (442°), and so much trouble was then experienced in connexion with the lubrication of the working surfaces and the packing of the glands, that in spite of its great economy this high pressure of steam was discontinued. Since that time much better lubricants have been discovered, and metallic packing has been introduced for the glands, so that now 315 lb. can be not only easily generated by many kinds of boilers, but safely and easily used.

Following the same line of illustration that we used with the steam of 160 lb. pressure, we will take the same cylinder of 3 feet area and see what theoretical result we can obtain by expanding the steam 27 times, and exhausting into a vacuum with only 2 lb. pressure. As we are cutting off at $\frac{1}{3}$ in each cylinder of our triple-expansion, we cut off at $\frac{1}{27}$ in the single cylinder. Its area is 432 inches or 3 feet, and the speed of piston at which we work is 600 feet per minute. If we filled one cylinder with steam at each stroke we should of course use 3 times 600 or 1,800 cubic feet per minute, but as we cut off at $\frac{1}{27}$ it will be $1,800 \div 27$, i.e. 66·66 cubic feet of steam per minute or 4,000 cubic feet per hour. The specific volume of steam at 315 lb. pressure is 85·7. Then $4,000 \div 85{\cdot}7 =$

46·67 cubic feet per hour, and 46·67 × 62·5 lb. = 2,926 lb. of water per hour. The power available will be as follows:— There are 432 square inches on the piston, the average pressure is 50·13 and piston speed 600 feet per minute. Thus $\frac{432 \times 50{\cdot}13 \times 600}{33{,}000} = 393{\cdot}7$ H.P., 2,926 lb. of water ÷ 393·7 H.P. = 7·4 lb. of water per horse-power per hour.

Doing this in one cylinder is of course quite out of the question owing not only to the extremes in temperature, but also to the intensity of the strains. For instance with a piston of 432 square inches and 315 lb. pressure on each square inch, we should have an initial strain of 315 × 432 = 136,080 lb.,

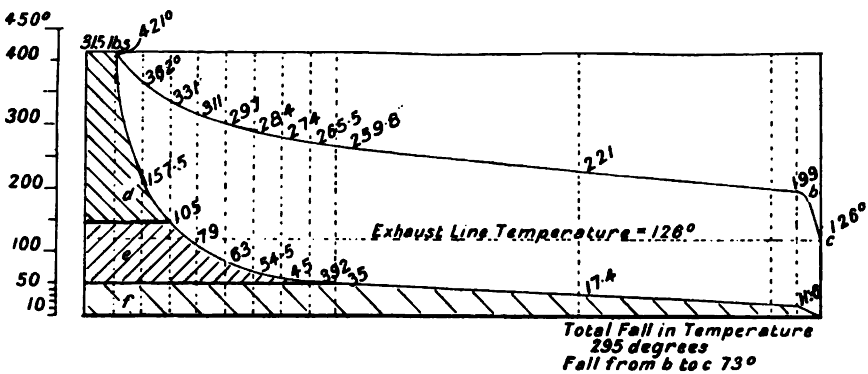


Fig. 39.

whereas if we use three cylinders, our total initial strain distributed over the three is but ¼ of this amount. The sizes of one cylinder to be proportioned to 27 expansions would be low pressure 3 square feet, or 432 inches; the second 144 inches and the high pressure ⅓ of this area, or 48 square inches. Then the initial pressure on each will be—

48 × (315 − 105) or	210	= 10,080	lb.
144 × (105 − 35) or	70	= 10,080	,,
432 × (35 − 2) or	33	= 14,256	,,
	Total	34,416	,,

Thus while the sum of the strain on each cylinder is but ¼ of what they would be in a single cylinder, the strain on any one cylinder and working parts is only $\frac{1}{9}$.

As in the case of the two-cylinder compound, the power given out is the same when the pressure and expansion is fixed, whether the steam be used in one cylinder or more.

It will be interesting at this point to look at the single-cylinder diagram, and the high pressure, H.P., intermediate, and low pressure diagram of the triple-expansion engine, and notice also the temperatures curves from each.

Fig. 39 shows the single-cylinder diagram, and it will be noticed that, notwithstanding the rapid fall in pressure at the beginning of the diagram, the temperature line does not fall so rapidly; second, that for nearly ¾ of the length of the stroke the fall in temperature is very small, but that at the opening of the exhaust valve there is a rapid drop from 199° to 126°, owing to the cooling action of the condenser, the

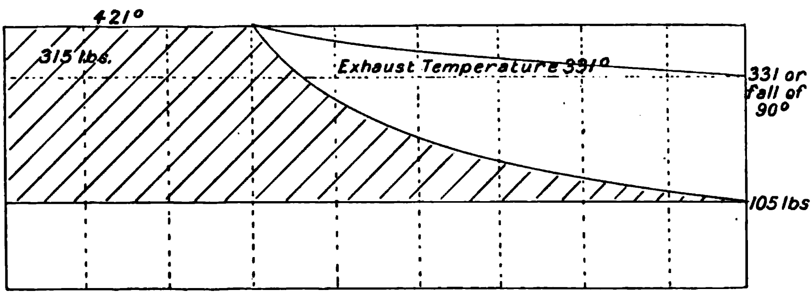


Fig. 39a.

cylinder being exposed to the temperature of the condenser during the whole of the exhaust stroke, which is half the time the engine is working; the total fall in temperature is 295°.

Fig. 39a shows a diagram from the high pressure cylinder of a triple-expansion engine working with the same pressure and number of expansions. We cut off in this cylinder at ⅓ of the stroke, the temperature remaining constant for the first third, from which point it gradually falls from 421° to 331°, or a drop of only 90°; 331° remains the temperature to which the cylinder is exposed during its exhaust stroke, which is shown by the dotted line.

Fig. 40 shows the diagram from the middle or "intermediate" cylinder as it is called. Here as in the high pressure cylinder, the initial temperature is carried on for ⅓ of its stroke and the fall from this point is very small, being only

71·2° for the whole of the remainder; the exhaust temperature is still high, so that there has been but little cooling action up to this point.

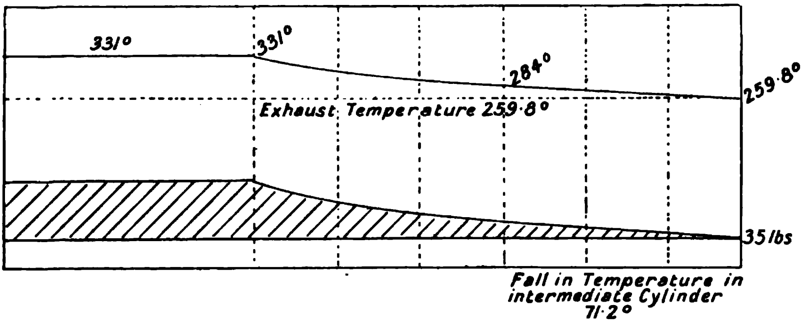


FIG. 40.

Fig. 41 shows the diagram from the low pressure cylinder; again the temperature remains constant for ⅓ of the stroke at the same temperature as the exhaust from the intermediate cylinder, then it falls very gradually until the exhaust valve opens, dropping from that point quickly from 199° to 126°. During the whole stroke up to the opening of the exhaust, the difference in temperature is only 61°, an additional 73° being due to the action of the condenser.

The upper line —·—·—·—·—·—·—·—·— shows what would be the temperature of the exhaust line were there

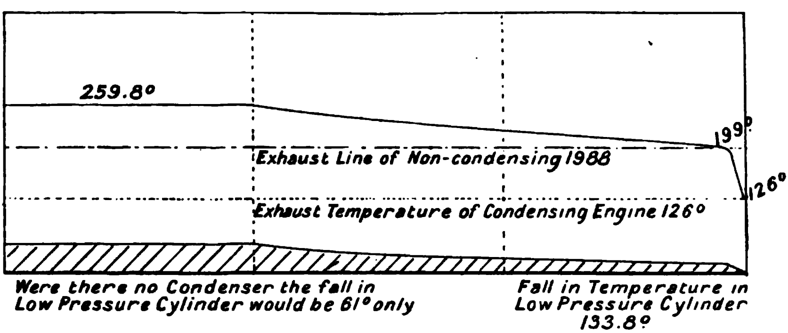


FIG. 41.

no condenser, and the dotted line below shows the exhaust temperature line with 13 lb. vacuum, the bottom line of diagram being the exhaust line were there a perfect vacuum.

The shaded portion in all the diagrams shows the steam pressure.

Combined or Composite Diagrams. These three diagrams can be combined by reducing the length of the high pressure and intermediate diagram in the ratio of the cylinder to each other. Thus, starting with the low pressure, we leave it as it is. The intermediate being $\frac{1}{3}$ of the area of the low pressure we reduce its length to $\frac{1}{3}$, and the high pressure being $\frac{1}{9}$ of the area of the low we reduce it to $\frac{1}{9}$ of its length, the heights in both cases remaining as they were. When this is done, the three together will form the same diagram, as if taken from a single cylinder. This diagram is shown us in Fig. 39, where the thick black lines show the division

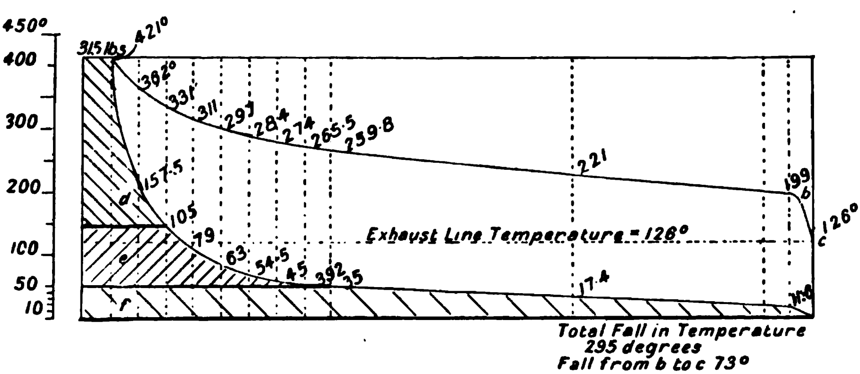


FIG. 39.

between the three diagrams, the high pressure *d*, intermediate *e*, and low pressure *f*.

These are manifestly theoretical diagrams, such as would be taken from an engine when the steam did its work without losing any of its heat beyond that expended in work. In practice we try to get as near to these as we can, and we shall show later, when we treat of the "indicator and valve setting," how near we can get.

It must be stated here that such an economy in the use of steam as we have shown to be theoretically possible, viz. 1 H.P. for 7·0 lb. of steam per hour, has never been reached and probably never will be reached in actual practice, though a consumption as low as 10 lb. is quite possible with 300 lb. pressure of steam if properly used. A large triple-

expansion condensing engine designed by the author (illustrated on p. 298), though only working with 220 lb. per square inch, when tested six months after setting to work, was found to use only 10·5 lb. of steam per horse-power per hour. With 300 lb. pressure an even greater economy would be reached. In connexion with the engine every cylinder was well jacketed with steam at boiler pressure in the H.P. and intermediate, and at receiver pressure in the low pressure cylinder, the receivers were also provided with heating coils. Nevertheless, when all the water of condensation from jackets and receivers was taken into account it was found that the engine was slightly more economical in the use of steam when the steam was shut off from the jackets, though the engine seemed to work more comfortably when they were used, doubtless owing to the re-evaporation of condensed steam in the low pressure cylinder.

Factors Making for Economy. Up to this point we have found, other things being equal, that, in order to use steam economically, i.e. to get the most work out of it—

It must be of high pressure.

That the piston speed must be high.

That it must be worked expansively.

That whether the expansion of steam at a given pressure be carried out in one cylinder or more, makes no difference to the power or economy which can theoretically be got out of it.

That when a sufficiently high pressure of steam is divided among two or more cylinders certain losses, due principally to condensation, are eliminated, and that in practice much higher economy is obtained.

That condensers attached to steam engines are not all gain, but are very advantageous in many circumstances, and almost essential in some.

That it is of vital importance to prevent all losses of heat due to radiation and conduction.

That steam jackets, though very convenient and useful, do not necessarily decrease the weight of steam used, but do conduce to ease and comfort in working.

In the next chapter we shall deal with the means employed in order to secure the best results.

CHAPTER VI

Valve Gears

We have now to consider how the conclusions we arrived at in the preceding chapters can be carried out in practice. We have dealt with the steam as it acts inside an engine cylinder. In this chapter we shall consider how it can be best got into the cylinder and out again when it has done its work.

In the beginning of the steam engine's history this was done in a very primitive way, simply by having two ordinary cocks; one was in the communication between the boiler and the cylinder and the other between the cylinder and the atmosphere or condenser, and it was the duty of the steam engine attendant to open and close these cocks as required, each stroke of the engine. When steam pressure was low and thus the cocks easy to turn, and the number of strokes was but four or five in a minute, this was possible to be done, though the work was not very easy. Tradition tells us that the first departure from this primitive method was in the case of an engine boy named "Humphrey Potter," who preferred playing at marbles to turning these cocks, and so fixed a cord to the beam of the engine in such a way that the engine opened and closed its own cocks while he enjoyed himself in play. We will give all honour to that boy, for all useless and unnecessary labour is foolish. This, however, is far from being a universal opinion. The author has a very vivid recollection of the time when a self-acting lathe was in few, if any, engineers' shop, what was known as a compound slide rest bolted fast to the lathe bed, and with the tool held in a short slide moved by turning a handle, was looked upon as a great advance, as it really was, over a tool held by the hand. An ingenious engine turner who worked near the author devised a simple means by which at every turn of the lathe

Humphrey Potter.

the handle of the rest was moved a little; it worked perfectly, and the man could fold his arms and joyfully look on, as he did, but not for long. He soon called down on his head a severe rebuke from an irate foreman, " You're paid your wages for turning that handle," he said, " and by —— you've got to do it or clear out; I'll have none of your —— laziness here"; and the apparatus was destroyed, to reappear not long after, when the old foreman had gone to that place where ingenious workmen will trouble him no more. The following year saw the birth in the same shop of the pioneer multi-tooled lathe and the split conical clutch which makes automatic turning possible. Anyway, Humphrey Potter's string was the beginning of self-acting valve gears. This was in the year 1713, and a few years later Henry Beighton improved it by suspending a rod from the beam, which was called a plug tree. From this was developed what is known as the cataract motion for vertical pumping engines, an exceedingly clever apparatus which is still used for some slow speed beam pumping engines.

The Slide Valve. The simplest method of admitting and discharging steam from an engine cylinder, is what is known as the slide valve, shown in perspective; this, as will be seen from Fig. 42, is a flat plate of metal with a hollow or

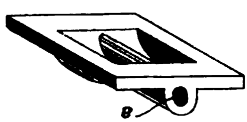


Fig. 42.

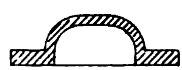


Fig. 43.

cavity in the centre of one of its faces; the cavity is shown clearly in Fig. 42, and the valve is shown in section in Fig. 43, and in orthographic projection in Fig. 42. The hole, as in Fig. 42, is for a rod to go through by means of which the valve may be drawn backwards and forwards on the cylinder valve face.

Fig. 44 shows a longitudinal and Fig. 45 a transverse section through a simple steam cylinder. In these figures *a* shows the slide valve in position on the working or valve face of the cylinder, *b* is the cylinder, and *c* and *c′* are passages or " ports " as they are called, leading from the valve face to either end of the cylinder, *e* is what is known as the " steam chest," and the

steam chest communicates, through the passage or pipe *f*, with the steam boiler; the steam enters in the direction of the arrow in *f*.

Piston-Rod.

In Fig. 44 *g* is a short cylindrical plug called a "piston" which fits steam-tight in the cylinder, and is free to slide from end to end; as shown, it is just ready to move in the direction of the arrows. A bar of iron or steel, *h*, turned perfectly smooth and parallel, is called the "piston-rod"; this is firmly secured to the piston *g* and travels to and fro with it, working through the "stuffing-box"

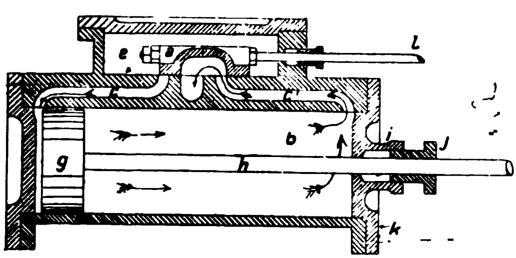


Fig. 44.

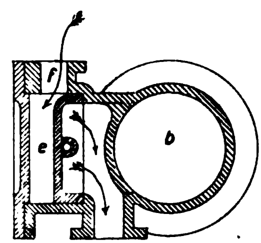


Fig. 45.

i and gland *j*; these make a steam-tight passage for the piston-rod through the cylinder cover *k*. Our object is to devise means by which the steam can alternately press upon the front and back of the piston, drive it backwards and forwards in the cylinder, and communicate its motion through the piston-rod to any mechanism outside the cylinder to which we wish to give motion; generally to make it turn a wheel by means of a crank.

For the present we confine ourselves to the cylinder and its valve.

If we look at Fig. 44 we shall see that when the valve chest *e*, is full of steam, and with the valve *a* is in the position shown, no steam can get out of the valve chest and into the cylinder, because all the ports are covered by the valve. If now the valve be drawn ever so little to the right (as it can be through

Steam Ports.

the rod *l*) the port *c* will be placed in communication with the steam in the valve chest, which, entering the cylinder in direction of the arrow, will first fill the small space between the piston and the cylinder cover, and then drive the piston forward in the direction of the arrows. As the piston moves forward, the valve also moves in the same

direction, and thus opens the port *c* still wider, until it becomes wide open, after which it begins to close again till, by the time the piston *g* has got to the other end of the cylinder, the valve *a* will have moved till the front edge is level with the port *c′* and the exhaust cavity covers port *c*; in other words to the same relative position with regard to port *c′*, that it occupies in the drawing with regard to port *c*. In the meantime the steam which had been used in a previous stroke to drive the piston to the position shown, and which now fills the cylinder, is free to escape through the port *c′* into the cavity of the slide valve, and from thence out through the exhaust pipe as shown by the arrows in Figs. 44 and 45.

Exhaust Cavity. The exhaust cavity in the slide valve *a* is always in open communication with the exhaust port and pipe, and by the sliding movement of the valve is alternately in communication with the ports *c* and *c′*.

The operation we have described is repeated at each end of the cylinder so long as there is steam in the steam chest; each movement, backwards or forwards, is called a stroke of the piston, and, of course, there are two strokes for each revolution of the crank.[1]

Crank and Connecting Rod. The way in which this reciprocating motion of the piston is turned into a rotating motion by a crank and connecting rod is familiar to all in the wheel of the travelling scissor and knife grinder, as well as in the ordinary foot lathe in which it has been used for many generations. In modern times the outside cylinder and crank pins of many of our railway locomotives are also familiar illustrations.

The following diagrams (Figs. 46 and 47) show in elevation and plan a simple form of modern horizontal engines where the essential parts are clearly indicated. The cylinder *a* is bolted on to the end of a trunk bed *b*, at the end of which is the shaft bearing *c*. On the end of the shaft *e* is the crank *d* firmly secured to it. On the end of the piston-rod there is fixed what is called the cross-head *e*, in the centre of which is the cross-head pin *f*. On the outer end of the crank *d* there is firmly fixed the crank-pin *g*, and connecting the piston-rod

[1] If no working model be available for showing this action a small one may easily be made out of cardboard or a ready-made one may be purchased at a low price from G. Phelps & Son; 32, Fleet Street, London.

with the crank through these two pins is the connecting rod *h*. It is evident that, as the piston with its rod and cross-head moves in the direction of the arrow (Fig. 46) it will push at the connecting rod *h* and turn the crank *d* round its centre. When

Dead Point. the piston reaches the end of its stroke, the piston-rod, the connecting rod and the crank will all be in a straight line, and however much the steam presses on the piston to bring it back again, it would be unable to move so

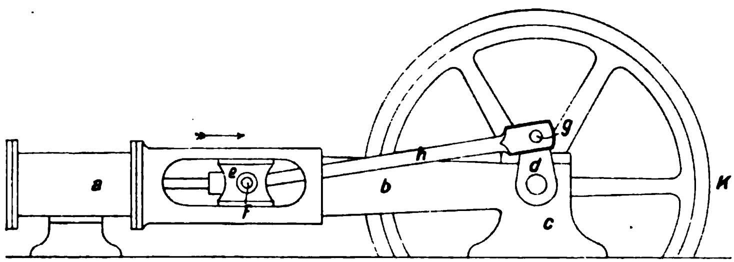


FIG. 46.

long as all remained in this straight line. This is called a "dead point" and occurs at each end of the stroke. To get over this difficulty what is called a "fly-wheel" is keyed on to the crank shaft (this is shown at *k* in Figs. 46 and 47). This fly-wheel, being rather heavy, continues to revolve after the piston has ceased to push it round, and thus carries the crank

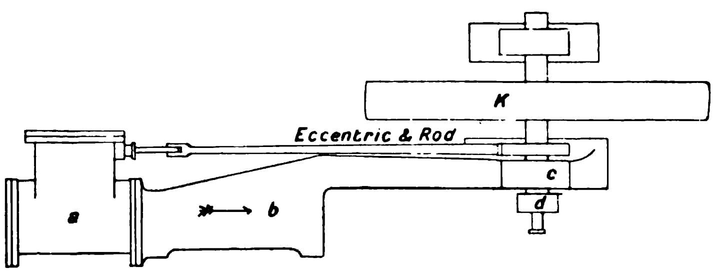


FIG. 47.

and connecting rod below, or above, the centre line or over the "dead point," as soon as it has passed which the piston can again move it.

What we now require is a simple means of moving the slide valve backwards and forwards just at the right moment.

This is accomplished by means of an excentric disc commonly called the excentric. This is shown in detail in Fig. 48, where *c* is the centre of the hole, or "bore" of it, which fits upon the shaft, and *d* is the centre of the excentric itself, its centre being "ex" or "out" of the centre of the crank shaft. The excentric is to all intents and purposes a crank arm of short length, the length of the crank arm in Fig. 48 being the distance between *c* and *d*, and the length of its stroke being equal to the diameter of the circle traversed by its centre shown by the dotted lines. Assuming the diameter of the circle to be 6 inches, then the length of the crank arm would be 3 inches, and the length of the stroke 6 inches. The crank-pin has to be of very large diameter, including within itself the shaft upon which the excentric is fixed, and the whole of the excentric itself. The diameter of Fig. 48 is shown by

Excentric.

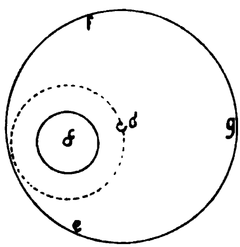


FIG. 48.

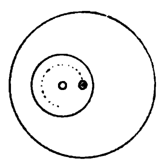


FIG. 49.

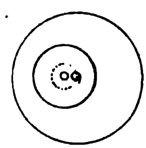


FIG. 50.

the encircling line *e*, *f*, *g*. The stroke of an excentric is called its "travel." Fig. 49 shows another excentric fitting on the same shaft, but with a smaller "travel," shown by the dotted lines within the diameter of the shaft and with a correspondingly smaller diameter, and Fig. 50 shows an excentric with still smaller travel and smaller diameter. It is excentrics of the proportions of the last, which are generally used on steam engines for working the valves, and the abnormal proportions of Fig. 48 is only shown that it might be clearly seen that an excentric is really a crank arm of short length and large diameter. If, therefore, one of these excentric discs be encircled by a hoop or ring accurately fitting it, and one side of the ring be fixed to a rod, the end of which is free to travel only in a straight line, the rod end will have a rectilinear travel equal to the travel of the excentric disc centre. Such an arrangement is shown in Fig. 51, where *a* is the crank shaft, *b* the

excentric or "excentric sheaf" as it is called, *c* is the surrounding hoop or "excentric strap," *d* the excentric rod, *e* the joint, and *f* the valve rod or "spindle" which is connected with the slide valve inside the valve chest. The position of the whole on a steam engine is shown in Fig. 47 marked "excentric and rod."

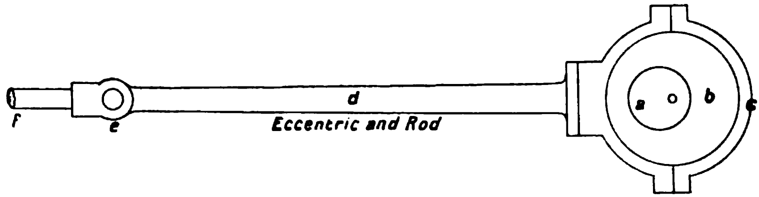


FIG. 51.

As excentrics for driving the valve gears of various kinds are almost universally used, it is worth while to spend some time over this really simple apparatus and also to clearly see the development of the slide valve from its simplest to its most complex and complete forms.

The first idea was merely to get the steam into one end and the other of the cylinder alternately, and some of the earliest valves were like Fig. 52.

Valve Working. It will be seen that this valve is just long enough to reach to, and just cover, both the steam ports *a* and *b* at once, any movement of it will therefore admit steam either into one end of the cylinder or the other. The inner cavity of the slide valve also just covers the space between the inner edges, so that this cavity is always in communication with the exhaust.

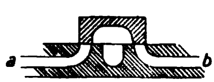


FIG. 52.

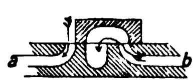


FIG. 53.

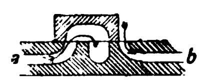


FIG. 54.

Supposing that the steam ports are 1 inch wide, then in order to open the port *a* the valve will have to move 1 inch to the right, and be thus in the position shown in Fig. 53, where steam is entering the port *a*, as shown by the arrow, the steam from the cylinder exhausting by the port *b*. Having attained this position the valve now moves to the left in its mid-stroke, becoming like Fig. 52 again, and, as it moves 1 inch more to

the left, reaching the position shown in Fig. 54, where the steam from steam chest enters by the port *b*, and the steam from cylinder exhausts by the port *a*. The total travel of the valve will then be 2 inches and the centre of its excentric will be 1 inch out of centre.

FIG. 52.

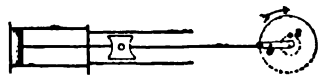


FIG. 55.

We will now see what relation this short crank or excentric has to the working crank of the engine. The small circle *a* with such a valve, is exactly at right angles with the working crank *b*, and this corresponds with position of valve in Fig. 52,

FIG. 53.

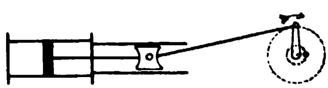


FIG. 56.

i.e. with the valve just ready to open. By the time the crank has got $\frac{1}{4}$ of the way round and the piston has made $\frac{1}{2}$ of its forward stroke, as in Fig. 56, the centre of excentric has got to the end of its stroke and the valve has got to position

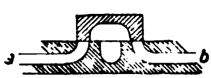


FIG. 52.

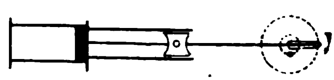


FIG. 57.

shown in Fig. 53, and the piston is being driven with full steam towards the end of its stroke; when it reaches the end, as in Fig. 57, the excentric crank is below the centre line, and has brought the valve back to its middle position, as shown

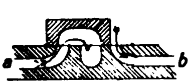


FIG. 54.

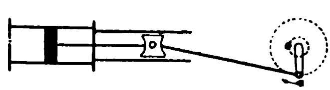


FIG. 58.

in Fig. 52, with the valve just ready to open port *b*, the piston being then at the opposite end of cylinder. From this point the revolution of crank continues till at $\frac{1}{2}$ stroke again, the piston is midway in cylinder and the excentric crank is at

the extreme end of its stroke, just ready to begin to close the port again. This is shown in Fig. 58. The valve has now got to position shown in Fig. 54, and the piston is under full steam going back to its first position, as in Fig. 55, the crank having now made a complete revolution, and the piston two strokes,

Fig. 52.

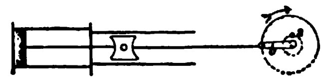


Fig. 55.

one forward and one back again. These operations will be repeated so long as we admit steam into the steam chest.

The illustrations given, above show how an engine would work with a primitive form of slide valve, probably the best form for very low pressures, but little, if any, above the atmosphere. We will now proceed to the development of the slide valve which still remains the simplest and most convenient form of valve which can be used for all pressures up to say 60 lb. above atmosphere.

Engine Diagrams. We have on several occasions referred to steam engine "diagrams," but in the former cases they have all been formed and drawn theoretically, though correctly, i.e. showing what would take place inside a cylinder when steam was being used under the theoretical conditions we have assumed. We have explained, however, that these theoretical conditions cannot yet be attained in practice, though we can approximate to them.

In order to see how near we can get to the ideal conditions we make use of a simple little instrument for measuring pressures at various points of the stroke of the piston, and thus indicate what actually is going on inside the cylinder. This instrument is called an indicator, and consists essentially of a small cylinder about half a square inch in area, fitted with a piston which is open on one side to the engine cylinder, and thus subject to the same pressure; this piston is held down in its cylinder by a strong but carefully tested spring of known strength. It is evident then that the indicator piston will rise and fall according to the pressure of steam in the engine cylinder. It can now be seen that if we attach a pencil to the piston-rod of indicator, this pencil could be made to draw a line upon a paper held convenient to it, and would

The Indicator.

mark on the paper the pressure within the steam cylinder. So far we have only got a straight line marked, but if we give a motion to the paper at right angles to the motion of the pencil, we can get marks in two directions, vertical and horizontal, i.e. if pressure remains stationary while the paper

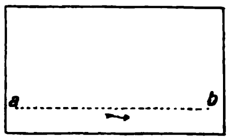


FIG. 59.

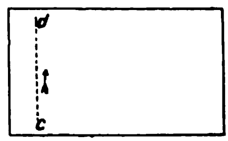


FIG. 60.

travels, we should get a horizontal line *a* to *b*, as in Fig. 59. On the other hand if the paper remains stationary while the indicator piston moved, we should then get a vertical line only, as *c*, *d* in Fig. 60, and hence it follows that if both are moving in equal times we should get the inclined line *a c*, to *b d*, shown in Fig. 61. From the above it will be evident that we may have any line formed from these two motions. Thus, if the indicator piston had risen more rapidly than the paper, then the inclined line in Fig. 61 would not have been straight but curved. What is necessary in using an indicator is, that the card should travel exactly in time with the steam engine piston. Starting when it starts, stopping when it stops (as it does stop at each end of the stroke) and moving always at the various rates at which the steam engine piston travels. A full detail and description of the various indicators and the best ways of using them will be given later,[1] but the above

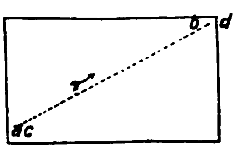


FIG. 61.

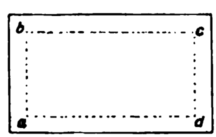


FIG. 62.

will be enough to enable the student to understand its principle and to know what is an indicator "card."

Now with such a valve as we have described, assuming we are working with a pressure of 60 lb. of steam, we should get a diagram which would be a parallelogram, as in Fig. 62.

[1] See Chapter XVII, page 343.

The moment steam was admitted, the indicator piston would rise, carrying its pencil with it, from *a* to *b*, the paper at the same moment begins to move with the advancing engine piston, and the line *b* to *c* would be drawn; at that point the end of the stroke would be reached, the steam would be shut off and the valve would begin to open the exhaust; this would cause the indicator piston to drop suddenly from *c* to *d*. Then the return stroke of engine piston begins, during the whole of which it is exhausting, and the indicator piston remains at the pressure of the atmosphere, making the straight line from *d* to *a*, thus completing one double stroke of one side of the engine piston. This is called an indicator diagram, and we shall often have to refer to them. The diagram we have got is a very simple but extremely wasteful one, and shows that we have thrown away into the atmosphere a cylinder full of high pressure steam which, though it has done a good deal of work, is capable of doing a good deal more. When steam pressure was only 1 or 2 lb. this waste didn't matter so much, but now that, as in our illustration, we are dealing with a pressure of 60 lb. above atmosphere it matters a good deal, as we want to make full use of all the pressure we have got.

How to do this with a simple slide valve is our next problem. We have learned in chapter IV the great value of expanding steam, by cutting off its admission into the working cylinder before the stroke of piston is completed; we want to do this with our slide valve if possible.

This can be accomplished by increasing the length of the slide valve by adding a piece at each end, *a* and *b*, as in Fig. 63.

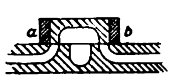


FIG. 63.

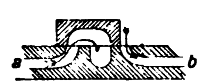


FIG. 54.

This valve is drawn to the same scale as Figs. 52 to 54, and

Valve Lap. the steam ports are again 1 inch wide. In Fig. 63 1 inch has been added to each end shown by the darkly shaded portions *a* and *b*. As with Fig. 52 we had a travel of 2 inches in order to open each port 1 inch, so with this valve, we shall now require a travel of 4 inches in order to give a full opening of the port, it having been found by long experience that the most suitable lap to put on a slide valve

is the width of the steam port. We may also state here, that when lap is put on a valve, thus increasing its travel, it is necessary to make the exhaust port at least double the width of the steam ports. In Figs. 52 to 54 the exhaust port is shown this extra width, but it will be noticed that in these the exhaust

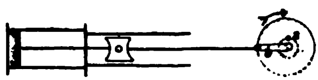


FIG. 55.

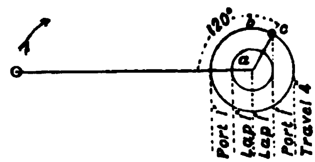


FIG. 64.

opening is always twice as wide as the steam port which delivers into it, whereas we shall soon see that owing to the greater travel with the "lap" on, the exhaust opening will be half covered twice in each revolution, hence the need for the extra width. Next, as to the relative position of crank and centre of excentric; formerly it was at right angles, as in Figs. 55 to 58. This position will no longer serve, and we must advance the excentric another inch, or the amount of the lap. There has been much unnecessary mystery made of the setting of slide valves, though the matter is really very simple. If a small diagram is made like Fig. 64 the matter will be understood at once.

Position of Excentric.

With the amount of the lap as radius, strike a circle *a*; with the lap and the width of port, strike from same centre a circle *b*; project upwards a line from the circumference of circle *a* till it cuts circle *b* and the intersection of the two at *c* will be the required centre of the excentric, if the crank-pin *d* is required to move in the direction of the arrow. If the crank is required to move in

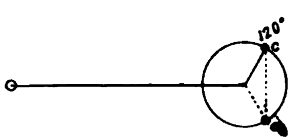


FIG. 65.

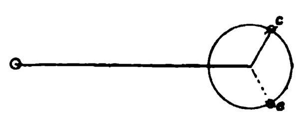


FIG. 66.

the contrary direction, then the line must be drawn downward to *e* (Fig. 65), which will be the position of centre of excentric. It will be noticed that we have drawn a diagonal line from centre of shaft to *c*. This line is exactly 120° from crank, and it cuts line *b* exactly at *c*. That being so, when the lap equals

the width of port, all that is necessary is to strike out a circle as in Fig. 65, giving full travel of valve, and with a 120° set square draw from centre till it cuts the circle. Here we get the same two points *c* and *e* for the forward or backward motion of crank that we did in Fig. 64.

A still simpler way of finding the centre of excentric (having struck out the line of travel) is, with the same radius, and with the centre line as centre, to draw curves intersecting the circle of travel, which it will do at *c* and *e* in Fig. 66, thus getting exactly the same points as we did in Figs. 64 and 65. It must be remembered, however, that this is true only when our lap equals the width of the steam ports, and this is an almost universal practice. When the lap is less or more, then the method pointed out in Fig. 64 holds good. For instance, let the lap be less, say ½ inch while the ports are 1 inch, then draw a circle *a* with ½ radius, as in Fig. 67, and another

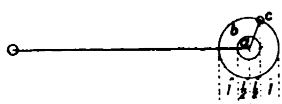


Fig. 67.

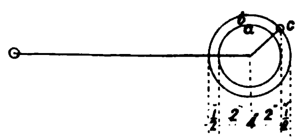


Fig. 68.

giving the travel, which will be ½ + 1 inch = 1½ inches. Project a line from *a* to *b*, and the intersection *c* will be the centre of excentric. Again, let the laps be more than width of port, say 2 inches and a port ½ inch, then with 2 inches as radius draw the circle *a* (Fig. 68), and outside it another, giving the travel *b*, which in this case will be a radius of 2 inches + ½ = 2½ inches giving 5 inches travel. From outside of circle *a*, project a line up or down, and where it intersects the line of travel, again will be the centre of excentric *c*.

It will be noticed that in the case of Fig. 67 the centre is further back than 120°, and in the case of Fig. 68 it is further forward (when we come to deal with "lead" we shall find it equally simple).

Having now fixed the position of the excentric when the lap equals the width of port, we will follow the action of the valve during one revolution; we will assume that the stroke of the engine is 16 inches; the path of the crank will be more

than this, as the crank-pin goes round a circle, while the piston travels in a straight line. In Fig. 69 the path of the crank is shown in dotted lines and is divided into three equal parts, A to B to C to D. The diagram (Fig. 69) represents the half stroke of the piston above the horizontal line, but it is evident that the return stroke must be the same as this. The crank is just starting its revolution in the direction of the arrow

Relative Motions of Piston and Valve.

from A to D. The position of the centre of the excentric is shown at E, and the edge of the valve in a line with excentric is just on the point of opening its port. In this and the five following diagrams, the valve is placed in an unnatural position under the crank, instead of in a line with it, as shown in Fig. 47; but by so doing

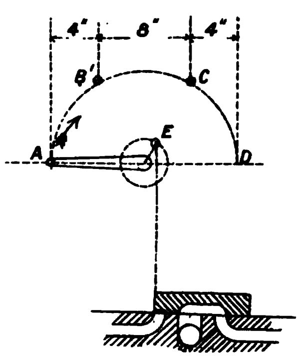


Fig. 69.

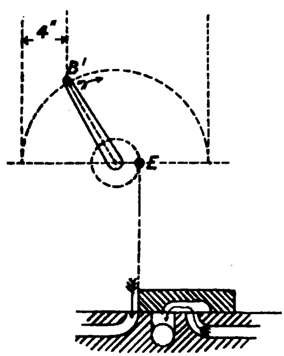


Fig. 70.

we can more easily see its direct relation with the position of crank-pin and centre of excentric. It will be noticed in Fig. 69 that while the crank moves from A to B, which is $\frac{1}{3}$ of its way to D, the piston, moving in a straight line, will only have moved 4 inches or $\frac{1}{4}$ of its stroke; further, we shall see that the excentric sometimes moves the valve at a rapid rate when the piston is going at its slowest, and at its slowest rate when the piston is moving at its fastest, which happens to be a great advantage to the working of the engine. Thus, in Fig. 70 the crank is shown having moved from A to B', during which time the piston has moved 4 inches only or $\frac{1}{4}$ of its stroke. By this time the port is wide open, the piston is moving now at its fastest, just double the speed at which it moved from A to B', and it travels another 4 inches to B'', as shown

in Fig. 71, while the valve has scarcely moved, only just beginning to close the port. From B' to C, another 4 inches, the piston moves equally rapidly, and the valve has started the rapid portion of its stroke, entirely closing the port by the

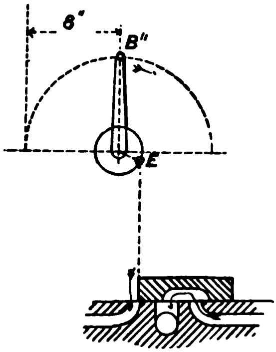


Fig. 71.

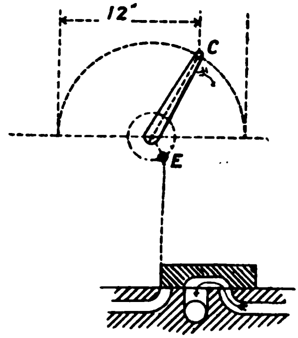


Fig. 72.

time the crank has reached C, as shown in Fig. 72. All this time while the crank is going from A to C the exhaust port, as shown in Figs. 69, 70, 71, and 72, is wide open, giving a free exit for the exhaust steam, and it closes very rapidly while the crank goes from C to C', as shown in Fig. 73.

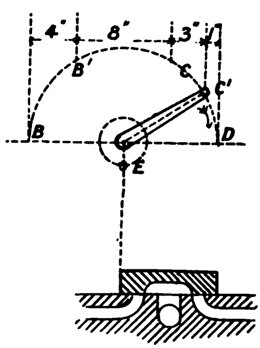


Fig. 73.

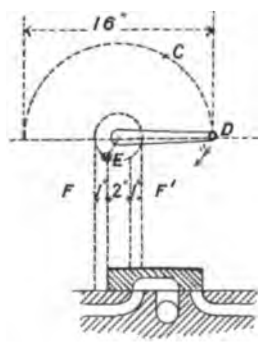


Fig. 74.

It is now seen why the exhaust port is made double the width of the steam port; though double the width, yet in Figs. 70, and 71 it is only about half open, and were it only the same width as the steam ports it would be entirely closed in Fig. 70, just when the piston is moving at its fastest, while, by giving it the extra width we have shown, it remains open for $\frac{15}{16}$ of the

stroke; this is clearly shown in Fig. 73, when the exhaust valve has only just closed. By this time the crank-pin has arrived at C' and the piston has travelled 15 inches out of 16 inches.

The piston at this point is moving very slowly, getting ready to stop and commence its return stroke. In Fig. 74 it has got to the end of its stroke and, while the piston has travelled that last inch, the slide valve has been moved the entire width of the steam port, and is now just ready to open for the return stroke, while the exhaust port is wide open to discharge the steam we have just used.

Having then seen the action of the slide valve during one stroke of the piston, we will now look at the action of the steam in the cylinder; this is shown in Fig. 75. From G to H represents the part of the stroke during which the steam is being admitted, 12 inches out of the 16. At H it

Cut-off.

is cut off, expanding from H to I, a distance of 3 inches. At I the exhaust port is opened, and the steam begins to escape, practically the whole of it (excepting just enough to fill the cylinder at atmospheric pressure) escaping

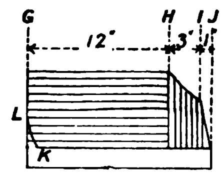


Fig. 75.

Exhaust.

at once into the atmosphere, making that loud "cough" so familiar to us in the locomotive. The pressure of the steam is still something, and if not free to escape, would exert considerable back pressure on the piston, thus retarding its speed; but with the wide exhaust port, and the disposition of the valve as it has been described, this back pressure is very small but still always something. Assuming it to be 1 lb. above the atmosphere, it would go on escaping as the piston returned, until it reached the point K at the bottom left-hand corner of diagram (Fig. 75) when the exhaust port

Compression.

closes, and what steam is left is compressed in the smaller space it has to occupy. Were there no clearance it would reach a very high pressure, the pressure increasing (as the space diminished), but there being always some "clearance," i.e. space between the piston and cylinder cover, and in the steam ports, this "compression" as we call it, seldom reaches to more than shown in the diagram from K to L.

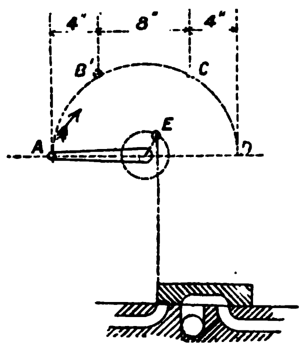


FIG. 69.

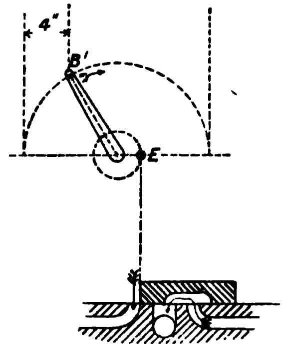


FIG. 70.

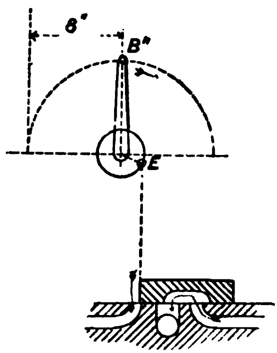


FIG. 71.

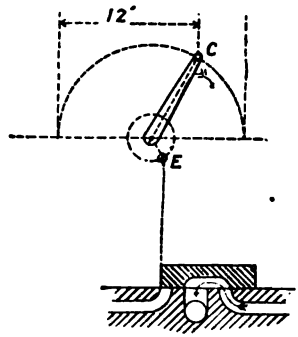


FIG. 72.

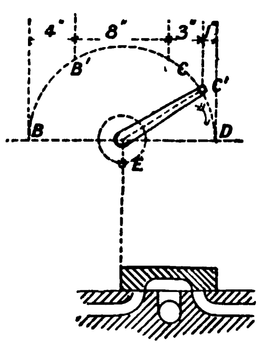


FIG. 73.

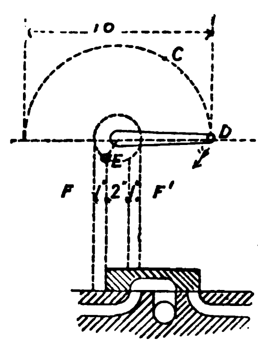


FIG. 74.

On page 96 all the positions of the valve, crank and excentric are shown at a glance, Figs. 69 and 70 showing the rapid opening of the steam port and the free exhaust; Figs. 71 and 72 showing the first gradual and then rapid closing of the steam port, the exhaust still being wide open; these four corresponding to the portion of the diagram marked in horizontal shading *G* to *H*, and representing $\frac{3}{4}$ of the piston stroke. Fig. 73 shows the position of valve corresponding with the part of diagram shaded vertically, and representing the expansion part of stroke, and Fig. 74 from *C′* to *D* showing the early exhaust from *I* to *J* occupying $\frac{1}{16}$ of the piston's stroke and the correponding compression from *K* to *L* also occupying $\frac{1}{16}$ of the stroke.

Now this expansion of the steam from *H* to *I*, though only for $\frac{3}{16}$ of the stroke is both useful and economical. Apart from the economy, such a distribution of steam as is shown in Fig. 75 conduces to easy working of the engine. The easy admission of steam, the expansion, the early and free exhaust, and the useful compression, making a spring cushion to stop the piston and help start it on its next stroke, are all valuable aids to the engine. We have now to consider how we can retain these advantages and get more economy by a higher grade of expansion, still with a simple slide valve.

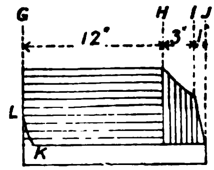


Fig. 75.

Advance. There are several ways in which we can do this, and among them, two good ways; one way is by altering the angle of advance with regard to the crank. We have seen in Fig. 64 how, when a lap equal with the port, the

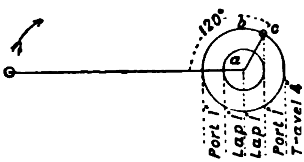


Fig. 64.

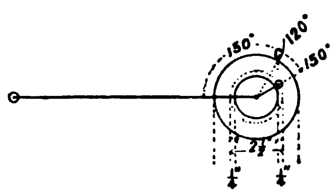


Fig. 76.

centre of excentric makes 120° angle with the crank. Let us now on the vertical line drawn from *C* to the lap circle find a point which shall be at an angle of 150° as in Fig. 76; by doing this we have reduced the travel of the

Reduced Travel.

excentric from 4 inches to $2\frac{1}{2}$ inches, and as the lap takes up 2 inches the port opening at each end will only be $\frac{1}{4}$ of an inch.

With this reduced travel we will now follow the action of the valve with relation to the piston as we did with Figs. 69 to 75.

In Fig. 77 the crank-pin is at *A*, and the centre of excentric at *C*, as described in Fig. 76; the excentric has $2\frac{1}{2}$ inches travel instead of 4 inches. In this first position we have the slide valve on the edge of the steam port just ready to open it, so that our lead remains constant. In the next diagram (Fig. 78) we find that by the time the crank has advanced 30°, or from *A* to *B*, we have got the widest opening of the steam port, the excentric being then on the centre line, and thus at the end of its travel. This opening is very small, but it suffices

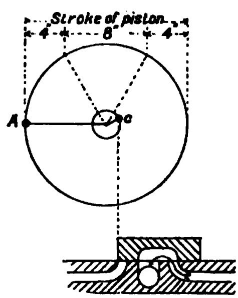


FIG. 77.

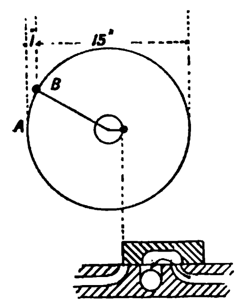


FIG. 78.

fairly well, for the piston is moving at its slowest rate, having only advanced 1 inch, or $\frac{1}{16}$ of its stroke, while the crank-pin has moved $\frac{1}{6}$ of its half revolution.

Going on to Fig. 79 we see that by the time the crank has made $\frac{1}{3}$ of its half revolution, the valve has closed the admission port again, and no more steam is admitted into the cylinder; by this time the piston has made but $\frac{1}{4}$ of its stroke, and thus the steam is said to be cut off at $\frac{1}{4}$. Both steam ports now remain covered until the stroke of the piston is finished.

It will be seen that in Figs. 77 to 79 there is a very clear and full passage for the exhaust steam, as shown by the arrows; but in Fig. 80, when the crank arm has reached *D* the slide valve is in a position just midway in its travel, and is in the act of closing the exhaust port to the right, and opening the one to the left, i.e. to the steam side of the piston. This occurs

while the piston is still 4 inches from the end of its stroke, and the effect is to cause a drop in pressure on the steam side of the piston and a gradual increase of pressure on the exhaust side, as the steam contained in the cylinder cannot now

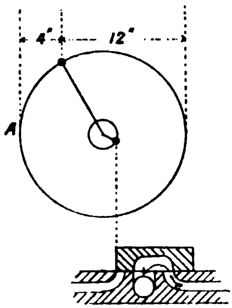


Fig. 79.

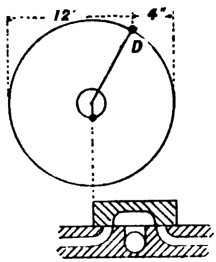


Fig. 80.

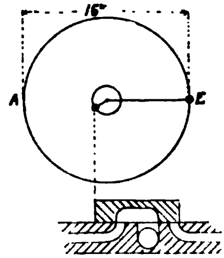


Fig. 81.

get out. By the time the crank has made its half revolution from *A* to *E*, the excentric has moved the slide valve to the position shown in Fig. 81, ready for the return stroke, when the above operations are all repeated.

The effect of all this inside the cylinders is shown clearly in diagram (Fig. 82). *A* to *B* shows the admission of the steam at full pressure, *B* is the point of cut-off, *B* to *C* is the same steam expanded according to the law explained on p. 46.

The expansion line is continued in dotted line to *D* where it would end but for the early exhaust shown in Fig. 80 and

Wire-Drawing.

which takes place at *C*. From this point there is a drop as the exhaust port is opened. Theoretically this should be a vertical drop, but, owing to the low pressure of the steam at that point and the very narrow opening out of which it has to come, the drop becomes a curved line *C* to *D*, at which point it reaches atmospheric pressure and can fall no more without a condenser. This line remains the same on the return stroke till it gets

Cushioning.

to *E*, at which point the exhaust port is closed and compression or "cushioning" begins, as shown by the line *E* to *F*. At *F* it joins the steam admission line, and another stroke goes on in the same order.

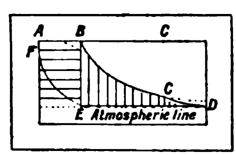


Fig. 82.

This diagram (Fig. 82) shows a good and economical dis-

tribution of a steam pressure of about 60 lb. above atmosphere, and, as a matter of fact, engines so fitted work extremely well. There are three drawbacks to this method of expansion, but they are not serious. First, the steam admission line from *A* to *B* should be quite horizontal as shown, but, owing to the small opening of the steam port and its closing again so soon, the line falls away as shown by the dotted lines. Next is the early exhaust while still 4 inches or $\frac{1}{4}$ from the end of the stroke; this again is not so bad as it seems, the pressure being low and the opening at the beginning very narrow, the steam line only falls slightly at first and more rapidly towards the end, when it soon meets the atmospheric line, so this objection is again slight.

The third objection is the excessive compression from *E* to *F*. This is but a slight one. True, this great compression means a corresponding loss of power, but not of steam, the compressed exhaust saving an equal amount of live steam from the boiler, and whatever force was used to compress it (as in the compression of a spring) is given out again when it expands.

All these defects are shown in a magnified form in Fig. 83, which is a diagram with a still higher cut-off, viz. $\frac{1}{8}$, then it will be seen that the expansion curve falls below the atmosphereic line at *K* before the opening of the exhaust, and would fall still further, as shown by the dotted line, but for the exhaust opening to the atmosphere, to which of course it at once rises; the compression is also proportionately greater. The horizontal shaded line shows in each case the volume of steam taken from the boiler (less the compression) to do the work. The vertically shaded part shows the gain from the expansion, which is seen to be very great.

Atmosphere Line.

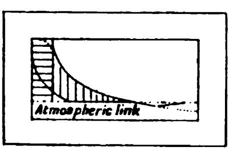


Fig. 83.

For the great majority of comparatively small engines used for agricultural and industrial purposes, often far away from cities and factories, and therefore worked by unskilled labour, such an arrangement of expansion gear is the best that has yet been designed; what it now needs is some means by which the point of cut-off can be easily altered and regulated.

One of the ways in which this is successfully done is shown in Fig. 84. In this Fig. *A* is the shaft upon which the excentric sheaf is fixed, *B* is a plate or disc firmly keyed on the crank shaft and through which the excentric is driven, *C* is the excentric sheaf. The circle *d* represents the travel of the excentric (in this case also the diameter of the crank shaft), *F* and *E* show the position of excentric centres at the angle of 120° from crank. Now as we have already seen, in order to keep the lead constant, while varying the point of cut-off, the centre of sheaf *C* must travel along a straight line at right angles with the crank from *F* to *E*. Now let *G* be a stud fixed into the excentric sheaf, and going through the plate *B* which has a slot *G G′* for the purpose. The excentric itself has a large slot *H H H* for the

Expansion Excentric.

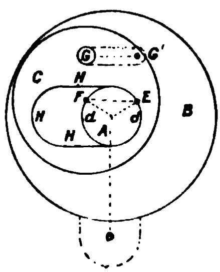


Fig. 84.

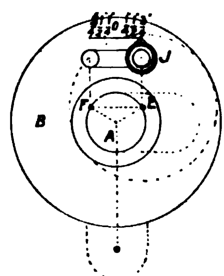


Fig. 84a.

purpose shown in Fig. 84 in full lines, and in Fig. 84a in dotted lines. The slot for the stud *G* is shown in dotted lines in Fig. 84 and in full lines in Fig. 84a. The stud *G*, which is screwed into the excentric sheaf, projects through the slot, and on the other side of the driving disc there is a nut with a pointed washer, shown at *J*. It is easily seen that by moving the stud towards the centre of the slot, we alter the angle of advance and the point of cut-off, the various points being indicated by the figures stamped on the driving plate and shown above the small slot in Fig. 84a. By moving the stud to the opposite end of the slot, the centre of the sheaf will be *E* instead of *F* (Fig. 84), and the engine will run in the reverse direction.

This ingenious arrangement was invented by a working man named "Chapman" in the employ of Messrs. Ruston, Procter & Co. about the year 1865, and has since been used regularly

by the firm, who paid him a royalty until the expiry of the patent. Another method of accomplishing the same object was designed by the author shortly afterwards and has been used since that time by the firm of Robey & Co., also at Lincoln. Both forms are used largely by many makers of engines in various parts of the world.

Figs. 85 and 85A show the front and back view of the

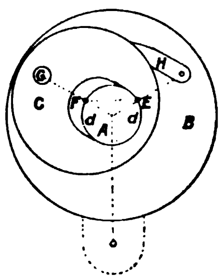


Fig. 85.

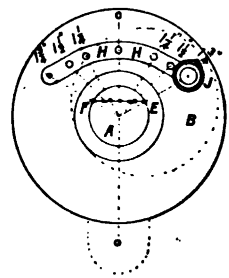


Fig. 85A.

adjustable excentric. *A* is the shaft as before, and *B* the driving disc keyed to the shaft. *F* and *E* are the centres of the excentric at its extreme forward and backward positions, and the straight line between the two, shows the line of travel of its centre, so that a constant line is maintained. *H H* is a curved slot in the driving disc (Fig. 85A), the small circles showing the various positions of the shaft centres in order to give $\frac{3}{4}$, $\frac{1}{2}$, and $\frac{1}{4}$ cut-off, the centre position of course giving no admission. With this arrangement it is only necessary to elongate the bore of the excentric sheaf, to an amount equal to the width of one steam port, the driving stud *G* is also placed in a direct line with centre line of excentric. Upon this stud, as seen in Fig. 85A, is a nut with a pointed washer, which can be set to the various points of cut-off, either in the forward or backward direction. The exact shape of the curve is easily got by dividing the travel line *F E* into any number of points, and with a distance equal to the distance between *F* and the centre of stud *G*, setting out this distance along the lines drawn through these points to the curve. If dots are made at these points, then the curve *H H* is drawn through the centres of these dots. In the example given in Fig. 85A, these lines are drawn at 30°, 45°, and 60°, and the centre of

the slot is practically straight between these points, connected by a segment of a circle. The point of cut-off in the cylinder corresponds to the position occupied by the stud *G*, at the various angles given, i.e. $\frac{3}{4}$ at 30°, $\frac{1}{2}$ at 45°, and $\frac{1}{4}$ at 60°, from the horizontal. The stud in this, and the former sheaf, can of course be set in any intermediate position, and the cylinder steam diagram obtained corresponds to Fig. 82 at $\frac{1}{4}$ cut-off with the corner rounded as shown by the dotted lines. Either of these methods work perfectly well, and there are approaching 100,000 of them in use.

Fig. 86 shows the method of reversing with a single excentric in use before the expansion plates we have described were invented, but while this makes an extremely simple reversing gear, it gives no range of expansion, being used always in its extreme positions. On the crank shaft, a stud pin is screwed exactly opposite to the crank-pin. In the boss of the excentric *C*, there is a slot shown in plan *d d′*; through this slot the driving pin *b* is screwed into the crank shaft; the length of the slot is equal to the curved distance between the centre of excentric at its forward and backward position and the diameter of the driving pin, $\frac{1}{2}$ at each end. The pin serves as driver, and when abutting against one end, the engine revolves forward, and when at the other end, backward. A set screw is usually fixed on the other side of the boss to hold the excentric sheaf from moving when not required. In our next chapter we shall deal with double slide valves.

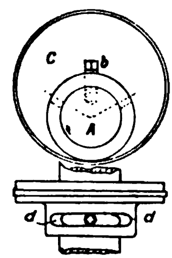


Fig. 86.

CHAPTER VII

Double or Compound Slide Valves

As stated on p. 100, the method of using steam expansively by a simple single valve is quite satisfactory up to pressure of about 60 lb. and a point of cut-off from $\frac{3}{4}$ to $\frac{1}{3}$ or $\frac{1}{4}$. For higher pressures needing greater expansion the slight defects pointed out become more serious. We have now to point out a way in which the slide valve can be used up to high grades of expansion without the evils of wire-drawing, early exhaust, or extreme compression. In order to accomplish this we make a further addition to each end of the slide valve, enclosing a part at each end as shown in Fig. 86A.

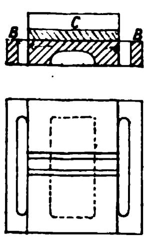


FIG. 86A.

The part between the arrow-heads is, like Fig. 63, including the lap at each end; the ports are each the width of the steam ports in cylinder; the bars B B are connected at the ends to the valve, and are more clearly shown in the plan.

Back Valves.

Upon the back of this valve is a plate C, called the "cut-off" or "back" valve, the length of this is equal to the width between the ports in main valve. This back valve is driven by a separate excentric placed on the same shaft as the main valve excentric, not keyed on the shaft, but driven by a bolt passing through both valve sheaves. If now both these two excentrics have the same travel, and are fixed at the same angular advance, the action will be just the same as a single valve such as was described in the last chapter, but if we advance the cut-off excentric to say 180° instead of 120°, which the main valve has, we get a very different action. Such an advance is shown in Fig. 87, and it will be noticed that the centre of the cut-off valve is exactly opposite to the centre of the crank-pin. That being so, the

cut-off valve will always move in the opposite direction to the piston, and their relative speeds will be the same.

In Fig. 87 the crank is just beginning its stroke, the main valve excentric *M* is opening the port, and at the same time the cut-off excentric *O* is moving opposite to the piston and

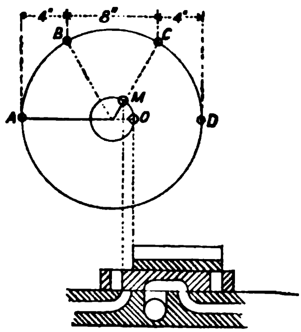


FIG. 87.

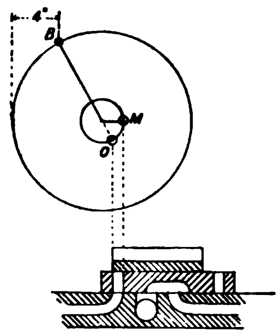


FIG. 88.

coming in the direction to close the steam port in valve. As they are fixed in the drawing, the steam port will have just got closed by the time the crank has made $\frac{1}{3}$ of its movement and the piston $\frac{1}{4}$ of its stroke as shown in Fig. 88. The port remains almost full open, till the last of this portion of

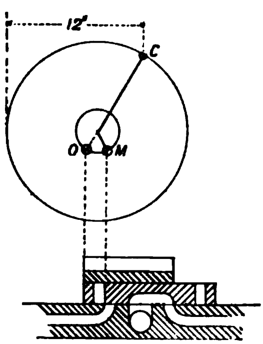


FIG. 89.

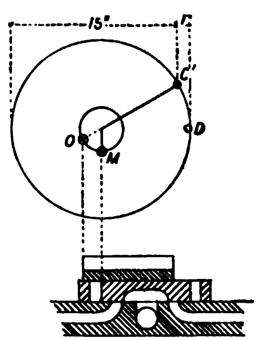


FIG. 90.

the stroke, the rapid closing being due to the fact that up to this point the two valves are moving in opposite directions. Fig. 88 shows the steam just cut off, and from that point, $\frac{1}{4}$ of the stroke, the expansion begins. The exhaust port at this point being wide open, and remaining wide open, until

the crank has got from *A* to *C* as in Fig. 89. In this Fig. it will be noticed that the two valves move in the same direction, the main valve *M* at a more rapid rate than the other. From this point the exhaust begins to close, and in Fig. 90 is shown just closed, but not until the piston has made $\frac{15}{16}$ of its stroke. From the position of *M* (*see* Fig. 90) it will be seen that the main valve is moving at its most rapid rate. so that by the time the crank-pin has got from *C′* to *D* the main valve will have moved sufficiently to the left to quite open the exhaust port to the left, and to be ready to open the steam port to the right, for the return stroke, when all the operations are repeated in the contrary direction.

Diagram showing Steam Distribution. With this setting of the valves we are cutting off at $\frac{1}{4}$. We will now look at the diagram we shall get, and compare it with the one we get from the single valve expansion at the same cut-off shown in Fig. 82.

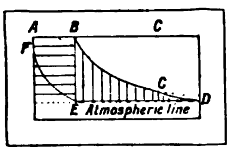


FIG. 82.

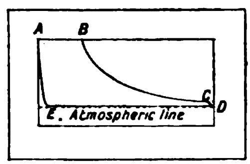


FIG. 91.

A is the point of admission, *B* point of cut-off, *B* to *C* is the expansion line, *C* the point of exhaust, *D* the end of the stroke, *D* to *E* the exhaust line, and *E* the commencement of the compression rising to *A* when it meets the entering steam.

Advantages of Second Excentric. This makes a perfect method of using steam of say 60 lb. above atmosphere with this cut-off, and when compared with Fig. 82 the difference is at once seen, and it is quite in favour of the second excentric and independent cut-off. Thus the admission is less throttled, the cut-off is sharper, the expansion is continued till nearly the end of the stroke, there is a freer exhaust, less back pressure and not so much compression. These are small differences and for expansion of $\frac{1}{3}$ to $\frac{1}{4}$ there is not much practical advantage, but there is some, and it is in favour of the back valve. With higher expansion the differences are still more marked in the same direction.

It is evident that if the cut-off excentric were keyed on to the shaft exactly opposite to the crank as described, there would (with definite lengths of valve) always be one point of cut-off, viz. $\frac{1}{4}$, as in Figs. 87 to 90. If, however, instead of this, the cut-off excentric were free to move forward and backward on the shaft, it will be able to be fixed in any required position.

This is accomplished by arranging the two excentrics as shown in Fig. 91A. Where *A* is the front view and *B*

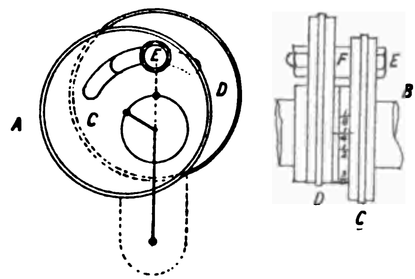


FIG. 91A.

the side view of the two excentrics, *C* is the main valve excentric and *D* the cut-off, *E* is a bolt going through the slots in the two sheaves. The bosses of the excentrics touch each other, one having a plain mark upon it and the other being divided, as shown, into spaces with marks for $\frac{1}{8}$, $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ points of cut-off. Between the two excentrics is the thick washer *F*, just filling up the space, so that when the bolt *E* is tightened it does not strain the excentric out of truth.

With this arrangement and the slots as shown, the cut-off excentric *D* can be brought back till it is coincident with *C*, or advanced still further in the direction of revolution till it cuts off at zero; thus there is the entire range between $\frac{3}{4}$ and zero.

Disadvantage of Fixed Position. The principal drawback to the arrangement is that it is necessary to stop the engine to alter the point of cut-off, and it is often advantageous, and always more convenient, to be able to make the alteration while the engine is running.

A little reflection will show that if (the cut-off excentric being fixed as shown in Figs. 87 to 90) we could alter the

length of the cut-off valve we could do this. When we wanted to cut off later, say at ¾ stroke, as in Fig. 89, we should merely want to make it 1 inch shorter at each end, then it would just cover the valve port at the same time that the valve covered the cylinder port. And conversely, if we

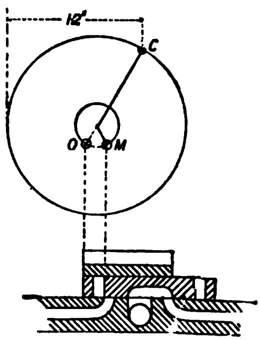


Fig. 89.

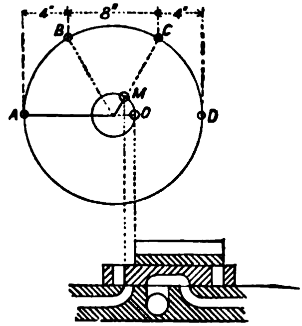


Fig. 87.

wanted to cut off earlier, say at half way between *A* and *B* (Fig. 87), by making the cut-off valve an inch longer at each end this would be done. This lengthening and shortening of the cut-off valve is accomplished by what is known as the "Meyer gear" ("Meyer"

Meyer Valve Gear.

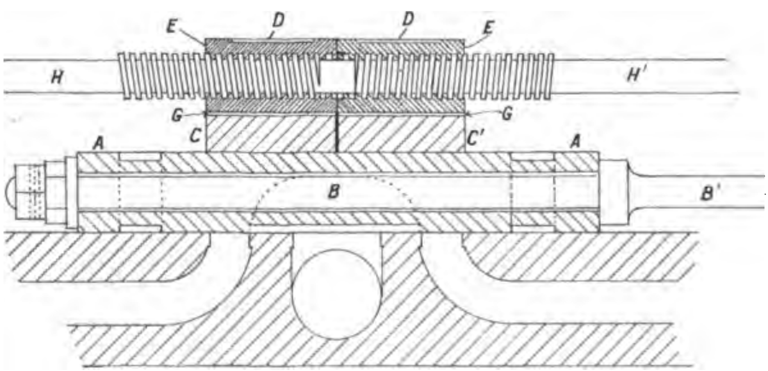


Fig. 92.

from the inventor); this is shown in detail in Figs. 92 and 93. This drawing should properly come where we treat of "engine details," but as it is advisable that the student should have a

very clear understanding of the Meyer gear, we give the valve detail now. *A* is the main valve, shown with its valve spindle *B* in position. At the end *B′*, this spindle projects through a stuffing-box at the end of the steam chest and connects with the main valve excentric on the crank shaft as shown in Fig. 47.

The cut-off or back valve is made in two parts, as shown, *C* and *C′*. On the back side of cut-off valve are projecting lugs *D*, *D*, *D*, *D*, within which are accurately fitted gun metal nuts *E*, *E* ; these, it will be seen in plan (Fig. 93), have flanges at each end at *F F* by means of which the valve is driven. These nuts are a good fit at the ends and sides, but are free to move a little up and down as shown by the clearance at *G*, *G*. The two nuts *E*, *E* have a square thread cut inside them, one right-hand and the other left-hand. The valve spindle *H H*, which gives motion to the back valve, has corresponding square threads right and left-hand cut upon it, fitting into the nuts.

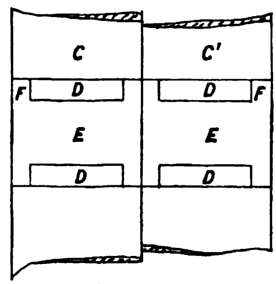


Fig. 93.

One end of the cut-off valve spindle *H′* projects through a gland at the end of valve chest next the crank, where it joins its own excentric rod, the connexion being made through a special joint which leaves the valve spindle free to rotate ; the other end *H* generally comes through the opposite end of the steam chest, on the outside of which it passes through the boss of a hand wheel, Fig. 98, within which it is free to move longitudinally, but it must rotate with the hand wheel. The action is now easily seen. On turning the hand wheel in one direction the two parts of the back valve will approach each other and thus the valve will be shortened, and, that by turning the wheel in the opposite direction, the two ports will recede from each other and the valve will be lengthened. Also, that this can be done while the engine is in motion, the hand wheel being fixed on a bearing, the spindle sliding freely within it, but compelled to turn round with it.

It will be seen on examining Fig. 89 that if the cut-off valve were shorter by the width of a port at each end, it would only just be cut off at $\frac{1}{4}$, so that all we have to do in this direction

is to shorten the valve to its utmost as shown in Fig. 92, and we shall cut off at $\frac{3}{4}$.[1]

The earlier points of cut-off require a little more consideration. With the same relative position of *M* and *O* Fig. 94

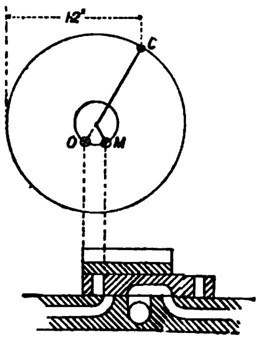


FIG. 89.

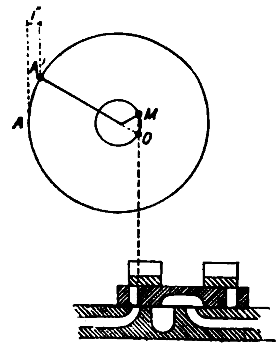


FIG. 94.

shows the cut-off valve extended until it cuts off at $\frac{1}{16}$ of the stroke. Here it will be seen that the main valve is moved to give a nearly, but not quite, full port opening, and that while the crank has only got from *H* to *A′* and the piston only moved $\frac{1}{16}$ of its stroke, the back valve has just closed the port. The reason why this occurs so rapidly is because

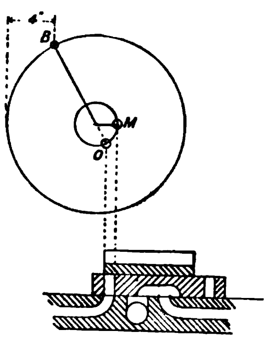


FIG. 88.

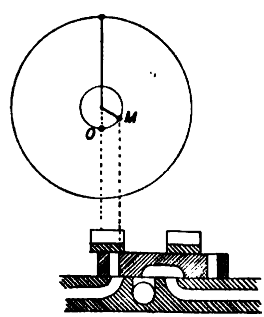
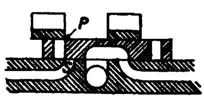


FIG. 95.

(as will be seen by looking at the position of *M* and *O*) the two valves are moving up to this point, in opposite directions.

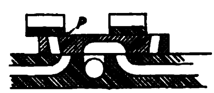
[1] By a different arrangement of excentrics and valves we could make it cut off later still—in fact, quite to end of stroke—but it is only in very exceptional cases that any later admission than $\frac{3}{4}$ is needed.

Shortly after, when the piston has moved another 3 inches, as in Fig. 88, M and O move in the same direction, but O at a much faster rate than M so that by the time half the stroke of piston is done as shown in Fig. 95 the back valve has got far in advance of the main valve and, as is seen from O and M, is still moving more rapidly. This goes on to such an extent that by the time the piston has made $\frac{3}{4}$ of its stroke the back valve has got so far ahead that it is just on the point of opening the steam port in main valve at its back edge as shown in Fig. 96 at P. By this time, however, the main valve has increased its speed till both valves have the same velocity. Further, at this very point the main valve has just closed the cylinder steam port at S, so the re-opening of the steam port is just avoided, and from this point the main valve moves more rapidly than the back valve.



1/16 cut-off

FIG. 96.



1/16 cut-off

FIG. 97.

Re-opening of Port.

The possible loss of steam from such a re-opening would, however, be enormous. The piston has got $\frac{3}{4}$ on the way to the end of the stroke, the steam inside has been expanded down below atmospheric pressure, when, through such a readmission, if made, the cylinder would be suddenly filled with high pressure steam to be thrown away again in the exhaust almost immediately afterwards. No other use of steam could be nearly so wasteful; much more wasteful than using steam without any expansion, as in this case we take a cylinder full of steam from the boiler almost at the end of the stroke of the piston and then get little or no work out of it. Numbers of times in the author's experience when investigating the cause of an abnormal consumption of steam, he has found it due to this cause, i.e. readmission.

As has been shown with valves properly set and arranged as in Figs. 94 to 96 the readmission could not occur, but the risk of it is so great that a careless or incompetent driver of an engine could soon bring it about. Therefore in order to remove the risk it is wise to incline the steam ports in the

main valve as shown in Fig. 97. They are sometimes splayed to the extent of half a port, and when that is done the danger point at *P* is entirely avoided even with so early a cut-off as $\frac{1}{16}$, which in practice is very seldom needed.

This form of valve gear is very largely used not only for the high pressure cylinder of an engine but also for the purpose of regulating the points of cut-off in the low pressure cylinder of compound steam engines, even when an automatic cut-off is used for the high pressures.

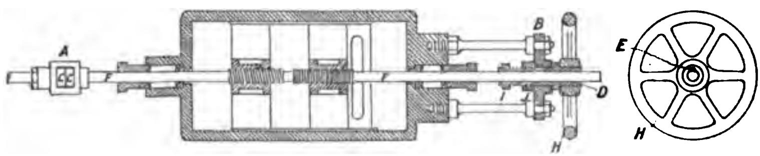


Fig. 98.

Regulating Gear. The method of regulating the Meyer gear when the engine is in motion is shown in Fig. 98. The steam chest is shown in section and the two halves of the back valve are shown in position on the main valve, widely separated as they would be for an early cut-off. One port in the main valve is shown uncovered.

The only new parts in this view are the swivel stirrup *A*, shown in detail in Fig. 99, and the crossbar, bush and handwheel shown at *B*, this also is given in detail in Fig. 100.

In the stirrup (Fig. 99) *A* is secured to that part of the valve

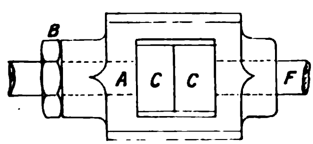


Fig. 99.

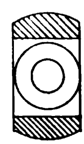


Fig. 99a.

spindle which goes towards, and receives its motion from the excentric on the crank shaft, and the stirrup is firmly locked on the spindle, *B* being the lock nut; the other end of stirrup has a plain swivel hole within which the spindle *F* can turn; the two round lock nuts *C C* are tightly screwed on to the end of spindle *F*, and, fitting accurately within the stirrup, serves to communicate its motion to the spindle and valve.

Fig. 99a is a transverse section through the stirrup. We

have now got a valve spindle in two parts but joined endwise by a swivel stirrup. One end of the spindle is driven in the ordinary way from the excentric, the other end carrying the valve is free to rotate upon its axis without affecting its length. The other end of the spindle *F* goes through the gland and out at the back end of the steam chest (Fig. 98), where it continues through the bush *D* ; on this end of the spindle and in the bush a square groove is cut and a key is fitted into the bush shown at *E* in the end view of the hand wheel. Upon this end of bush the hand wheel *H* is keyed, so that when the wheel is turned the bush and the spindle turn with it.

This bush is turned smooth on that part which goes through the crossbar *B*, but the part projecting beyond the crossbar towards the steam chest has a screw thread cut on the outside ; the nut *J* is screwed on this part and pinned to the bush so that it turns with it and thus holds the bush in position endwise ; another nut *I* is screwed upon the bush, but this nut is held from turning round by the plate *K* (Fig. 100) through

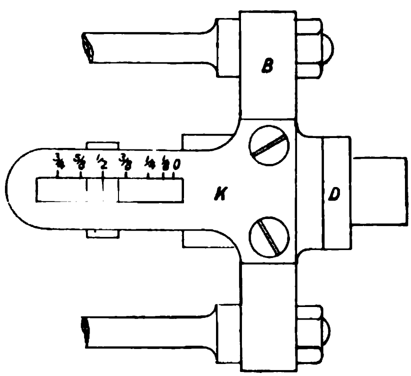


Fig. 100.

the slot in which a projection on the nut slides. As this nut cannot turn with the bush, it follows that as the bush is turned, the nut will be screwed backwards or forwards upon the length of the screwed part of bush, and its projection coming through the slot in plate *K* will slide from end to end of the slot. This sliding nut serves two purposes, it prevents the bush being turned too much in either direction, and an index mark upon it can be set to marks on the plate showing by its

position on the bush the various points of cut-off in the steam chest.

Automatic Meyer Gear. Many engineers have attempted to make the Meyer gear automatic by connecting the cut-off valve spindle in such a way to the engine governor that as the governor slide rises, the valve spindle is turned round. All these gears are more or less failures owing to the great friction of the parts and the number of revolutions which have to be given to the spindle. Students are cautioned, therefore, against wasting efforts in this direction, especially as, when we come to consider automatic valve gears and governors, we shall find so many simpler and better ways of doing what we want.

It will be seen from the foregoing how very simple and admirable an apparatus is the slide valve when well proportioned. We will now consider its defects and how to remedy them.

For small engines and moderate pressures the slide valve leaves nothing to be desired. Even for very powerful engines like locomotives, working with pressure of 200 lb., its simplicity and reliability are such advantages, that its defects are put up with in the majority of cases. The principal of its defects, and some would say its only one, is the large amount of power it takes to drive it when high pressure is used, on account of the friction on its working face.

Valve Friction. This friction also tends to a very rapid wear of the valve and the valve face and a consequent leakage of steam.

The friction is due to the great pressure on the back of the valve, one side of which is exposed to the full pressure of steam in the steam chest, while the whole of the exhaust cavity in the valve is exposed to little or no pressure, being in constant communication with the atmosphere or the condenser.

Taking the case of the valve we have used in our illustration and assuming that the ports are 10 inches long, then we have an exhaust cavity in the valve 4 inches wide by 10 inches long or 40 square inches in area. This with 80 lb. pressure would mean $80 \times 40 = 3,200$ lb. pressure on the valve due to exhaust cavity alone. The friction of two polished and lubricated metal surfaces is according to table, $\cdot 15$ of the pressure when at rest and $\cdot 7$ of that when in motion, then the pull on the valve will be $3,200 \times \cdot 15 \times \cdot 7 = 336$ lb. The valve moves 4 inches in each

direction or 8 inches per revolution of engine. If thus the engine makes 150 revolutions we shall have $\frac{150 \times 8}{12}$ or 100 feet of valve travel per minute $\frac{336 \times 100}{33,000} = 1$ H.P., a quite appreciable amount of power to be saved if we can; but more important than saving the power, is the saving in wear and tear of the valve face and valve gear.

It may reasonably be asked why, in calculating the pressure on the face of the valve we did not take its whole area, which is 96 inches, instead of the cavity which is only 40 inches? The reason is, that the exhaust cavity is all the time subject to exhaust pressure, while the other parts of the valve would be, but for this, nearly always in equilibrium. A solid block of metal with smooth surface could be pulled backwards and forwards in a chamber full of steam of any pressure without it causing any resistance, because the pressure would be all round like the air is all round our bodies, and therefore exerts no pressure on one side more than another.

Air Pressure. Most of us when boys, were familiar with the experiment of placing the neck of a small bottle in the mouth, sucking the air out of it, and closing the neck with the tongue. In doing this and thus removing the pressure of the air from a very small part of the body (only about ½ of a square inch) we find that the pressure of the outer air forces the surface of the tongue into the bottle and forces the bottle on to the tongue, so that if the vacuum be good it will take a pull of about 7 lb. to remove it from this unbalanced part of the body. Were it possible to expose one whole side of the body to a vacuum while the air pressed on the other side, it would result in a pressure of more than 6 tons.

In the case of the slide valve, not only the exhaust cavity but a portion of the flat part which covers the steam port, is exposed to a low pressure as the steam in the cylinder expands, but this is only for a comparatively short time, and even were it otherwise, it would not be safe to balance the valve entirely as it is most important that it should always be held to the valve face. A small difference in pressure is therefore advisable. It is evident that if we could remove as much pressure from the back of the valve as is naturally removed

from its face, we should place it in equilibrium so far as the exhaust cavity is concerned. This is done by exposing the back as well as the face to the same low pressure in the manner shown in Fig. 101, where *A A* is the valve in section and *B B* in plan. *C* shows the exhaust cavity which is 4 inches × 10=40 square inches in area. This corresponds to a diameter of 7⅛ inches, so that the short cylinder *D* on the back of valve is bored out to that dimension; within this fits the equilibrium ring *E*, the flange of which abuts against the steam chest cover *F*. For an amount equal to the diameter of the flange plus the travel of the valve, this cover has to be tooled and scraped perfectly true, so that an equally true surface placed upon it would make a steam-tight joint. It is also essential that its surface when in position should be exactly parallel with the valve face of the cylinder. There is no difficulty about fulfilling both these conditions in a well-equipped engineer's shop, and their neglect ensures failure. The equilibrium ring should be ground into its place and fit like a template just easy enough to move with a slight pressure; it is best also to leave it free to revolve if it wishes, as this promotes even wear. If it be a good fit, and three shallow grooves be turned in its circumference as shown, it will be practically steam tight even under high pressure. One other point is essential, the part of the ring flange *E* which slides against the surface of cover, must be about 10 per cent. less in area than the under part, then whatever be the pressure, it will tend to press the ring against the face of cover just enough to keep it steam tight. If these proportions and precautions are taken the valve will work almost without friction and with no perceptible leakage.

Valve Equilibrium.

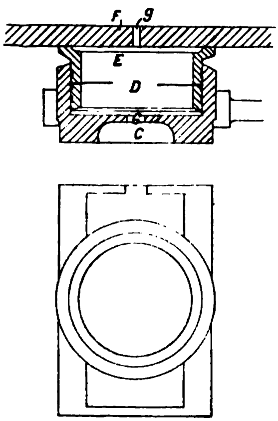


FIG. 101.

It will be noticed that a hole *G* is made in the cavity of the valve, thus putting the exhaust in free communication with the equilibrium ring and reducing the pressure within the ring to that of the exhaust. Another hole is made in the steam chest cover at *g*. This is fitted with a test cock. When

the engine is running, and the cock is opened, there is a slight escape of steam from it, not necessarily the result of leakage but corresponding to the pressure of the exhaust. Any leakage past the equilibrium ring can be easily detected by means of this cock which, though ordinarily closed, should be used occasionally for this purpose. In the case of a condensing engine a small pipe should be carried from *H* to the condenser, the pipe having a cock in it to be used for the purpose indicated above.

Object of Balancing Valve.

Another way of doing the same thing is to put the true plate on the back of the slide valve and put the equilibrium ring in the steam chest cover; this is shown in Fig. 102. The working action of this is exactly the same as Fig. 101 and the former construction is most frequently used.

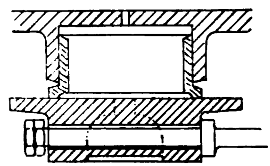


Fig. 102.

Balanced slide valves are, however, seldom used except in the case of the low pressure valves of large short-stroke engines and less for the purpose of saving fuel than for the diminution of the strains on the working parts of the valve gear.

Trick Valve.

Another method of reducing the friction by reducing the travel is by the employment of what is known as a "trick" valve; this is shown in Fig. 104. For the sake of comparison a standard slide valve is shown in Fig. 103. The following illustration of "trick" and double-

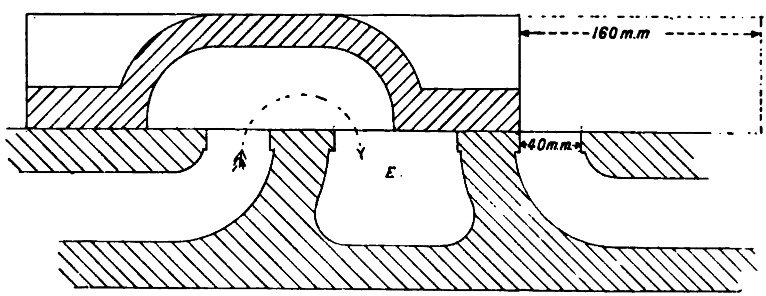


Fig. 103.

ported valves are drawn to the same scale, the ports are of a size which gives the same opening to the steam, the difference is in the "travel" of valve. Taking one steam port as unity,

then as is shown by the dotted lines the standard simple valve (Fig. 103) has a travel of four ports, i.e. the steam port being 40 mm. the travel is 160 mm.

Fig. 104 shows the trick valve. This valve has a passage

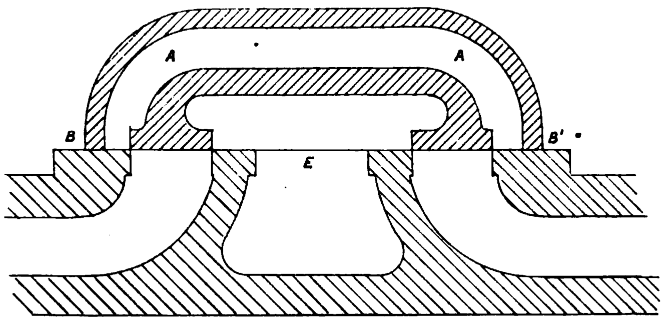


Fig. 104.

A A made within it, and it works on a cylinder face only slightly different from Fig. 103. The total width over ports is the same, but each steam port is made the width of *B* and *B′* wider, this being, because in opening the steam port, it is partly covered by *B* as shown in Fig. 105. The width of the exhaust port *E*, is however, less, as this port is not diminished to anything like the same extent as the standard valve

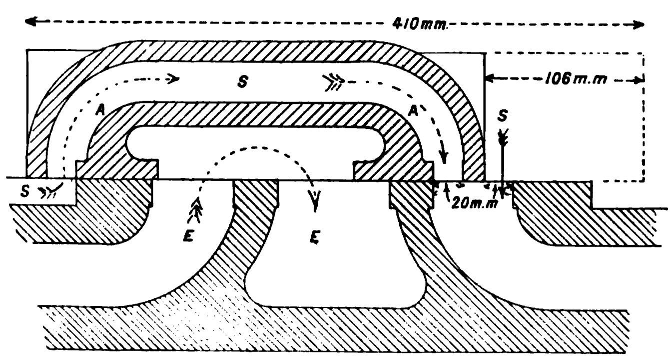


Fig. 105.

port (Fig. 103) which, as that figure shows, is reduced to ½ its width when the steam port is wide open. In Fig. 105 the trick valve is shown at one extreme of its travel, with the steam and exhaust ports wide open, the arrows showing the direction

of the steam. Here it is seen that though the outer edge of the valve only opens the steam port 20 mm. instead of 40, yet there is another opening of equal extent through the steam port *S* from the other end of the valve, this opening

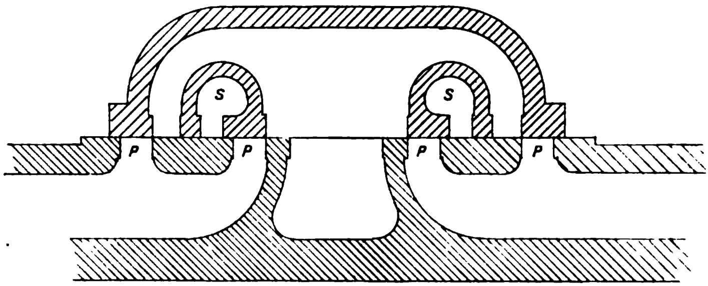


FIG. 106.

being made at the same time as the outer edge opening. This valve, as in Fig. 103, is shown at the full extent of its travel in one direction, the other extreme being shown by the dotted lines, and the amount of its travel being 106 mm. as against 160 with Fig. 103, and a total length occupied in the steam chest by valve and travel of 410 against 486 mm.

Double-Ported Valve. A still greater saving can be made by the use of a double-ported valve and double-ported cylinder face as shown in Fig. 106.

In this case instead of one steam port of 40 mm. we have

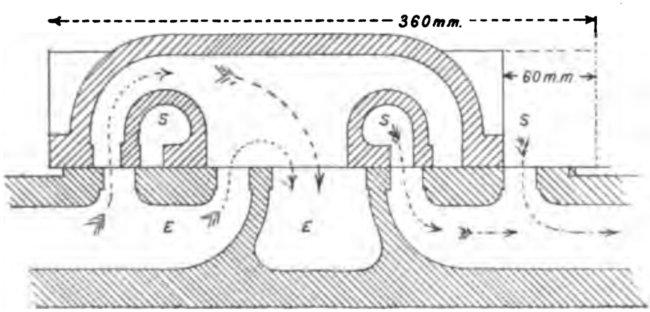


FIG. 107.

at each end two ports of 20 mm. each as shown at *P P*, *P P* with a common exhaust port *E* in the centre. There are practically two branches to one port. The proportions of all the parts are clearly shown in Fig. 107, where the valve is shown

at one end of its travel with steam and exhaust ports wide open, the direction of flow of the steam being clearly shown by the arrows *E E* being the exhaust and *S* the steam ports. This valve gives the shortest travel of all, the dotted line in Fig. 107 showing that the total travel is only 60 mm.

The valve, however, is of rather complicated make, and the ports of cylinder are difficult in moulding, extreme accuracy also is required in fitting up. The use of such a valve is only justified in large engines of very short stroke.

All these valves can be made to work in equilibrium by adopting the methods described in Figs. 101 and 102.

Double-ported and trick valves do not lend themselves to be used as expansion gears with Meyer motion, but as single valves with points of working cut-off, of from $\frac{3}{4}$ to $\frac{1}{8}$ they are

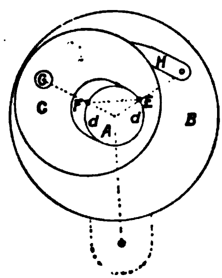


Fig. 85.

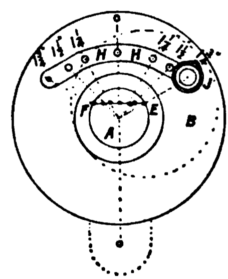


Fig. 85a.

very useful. It is evident that, like other single valves, they could be worked with adjustable excentrics as in Fig. 85, and this is what is frequently done. When, however, it is required to adjust the point of cut-off while the engine is in motion other methods have to be adopted. A simple way of doing this, designed by the author for use in connexion with large vertical engines for working dynamos direct in electrical stations, is shown in Fig. 108.

How to Adjust when in Motion.

This is a trick valve like that shown in Fig. 105. It is worked by two excentrics, one giving it the full travel and being at an angle of 120° to the crank, the other being set opposite to crank, or at 180° and giving a shorter travel, to the edge of steam port only. The ends of the excentric rods are connected together by a link. At the extreme ends of the

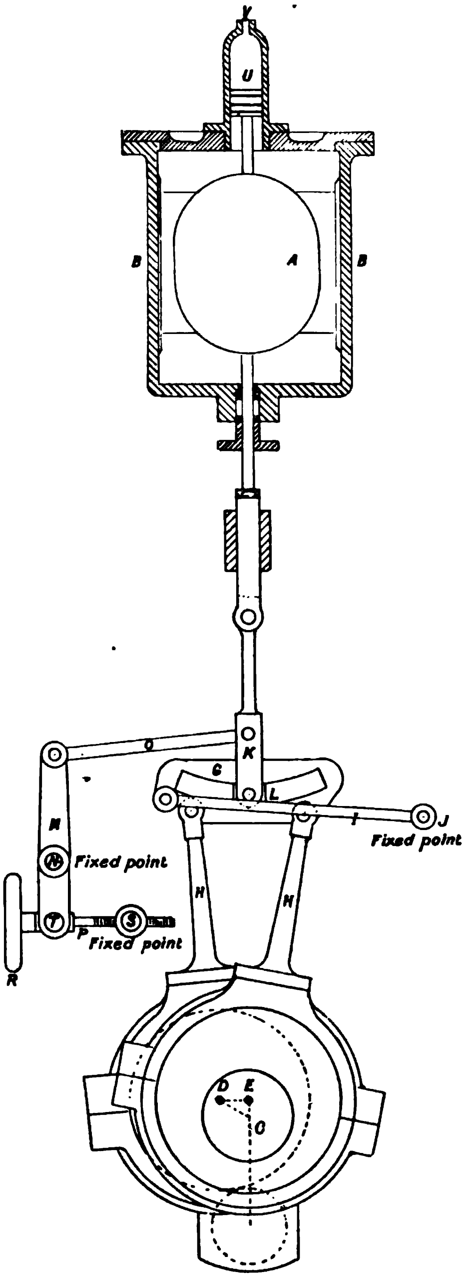

V
U
B
A
B
O
K
G
L
J
I
Fixed point
M
N
Fixed point
H
H
T
S
P
Fixed point
R
D
E
C

FIG. 108.

link, either one or the other of the excentrics has full control of the valve motion, the other excentric working idly, but at any intermediate point both excentrics control the valve, and as the link is brought from one to the other the change gradually takes place. We have thus the power to alter the angle of advance from 120° to 180°, and at the same time to change the cut-off from $\frac{3}{4}$ to zero, the lead all the time remaining constant.

In Fig. 108 the curve in the slot link *G* is struck from the joint of the radius rod *K*. So the link is fixed so far as travel in its length is concerned. It is evident that the link could be moved lengthwise, till one excentric rod or the other comes directly under the valve spindle. It is, however, more convenient and easier to move the radius rod *K*.

A is the valve with its equilibrium plate on the back, working in the steam chest *B*. *C* is the crank shaft with its two excentrics with centre respectively at *D* and *E*, *D* giving the full travel and *E* the smaller travel and cut-off. *G* is the link connected to the two excentric rods *H H*, and held in position by the radius rods *I*, only one of which is seen, the other being in the same place on other side of link.

They work on the fixed centre *J*. *K* is the valve spindle radius rod, a sliding block *L* working in the curved slot in link *C* forming its connexion with the excentric rods. For moving the radius rod a lever *M* is used, working on the fixed point *N*, one end of the lever being connected to the lifting rod and the other to the screwed spindle *P*. On one end of the spindle is the hand-wheel *R* and the other end works in a nut *S*. The nut and the bearing of the spindle *T* being cylindrical can accommodate themselves to the various positions the spindle has to take, the end of lever *M* moving in a curve. (We shall see later how the radius rod *K* may be controlled by the engine governor when the variation of the cut-off would be made automatic.)

Balancing Cylinder. Such valves as these, especially when they have balance plates on the back, are of considerable weight, and when they are placed vertically, as in Fig. 108, it is advisable to balance the weight as well as the pressure on them. This is done in a simple way by using a small cylinder *U* on the top of the steam chest fitted with a piston which is connected with the slide-valve spindle, thus balancing the

weight of valve. Of course the size of cylinder must be proportioned to the weight of valve and the average steam pressure in the valve chest. A small tube should be connected at *V* to take away the leakage, if any.

Cock Valve. A form of valve which has often been tried, but seldom with satisfactory results, is the plug or cock valve shown in Fig. 109. Here instead of a flat-surfaced rectilinear box, the steam chest is a small cylinder bored out at right angles to the working cylinder; the steam and exhaust ports are set out in the same manner as a flat valve. The valve itself, *A* in Fig. 109, is turned to fit well but easily in its

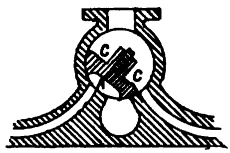


FIG. 109.

cylindrical steam chest and has a reciprocating motion given to it by a rocking lever worked by an excentric in the usual manner. This valve is to all intents and purposes a slide valve with a curved face. It has been thought by some that such a valve works with less friction than a flat slide valve, but a little reflection will show that this cannot be; the steam presses on its back at *c c* with just the same force, and the exhaust cavity is of just the same size as if it were a flat valve, and it has the disadvantage that the working edges of the ports cannot without much difficulty be tooled perfectly straight and true, as all port edges should be. The steam ports are also necessarily a little longer.

The above form of valve, though not specially good in itself, has helped to give birth to one of the best forms which it much resembles, viz. the "Corliss valve," as it is called from its inventor. In this form the curved slide valve is practically divided into four parts, each being a portion of a cylinder working within a cylinder. In the plug valve it is seen that there are four working edges to the valve, i.e. the two outer edges giving the admission and the cut-off, and the two inner edges of the exhaust cavity. In the Corliss gear each valve has one working edge only. This gear we shall deal with in our next chapter.

CHAPTER VIII

Releasing Valve Gear

By releasing valve gear we mean working the admission valve of a steam engine by a positive motion, at a given point in the revolution retaining it wide open as long as is required, and thèn suddenly releasing it at a subsequent time, which may be fixed or variable, afterwards letting it remain entirely at rest until the time comes for opening again.

Probably the earliest of these in point of time was the "Cornish drop valve and cataract motion." This, however, did not get much further than its application to slow-moving pumping engines, for which purpose it is still used.

Corliss Valve Gear.

The form of releasing gear which has been most widely used during the last fifty years is the "Corliss," briefly referred to in the last chapter. Fig. 110 shows a section through a Corliss cylinder with long stroke.

Corliss Engine.

It will be well at this stage, in order that we may better understand what follows, just to take a look at Fig. 125 (p. 136), which gives a general view of a Corliss engine. The engine has a bayonet trunk frame such as is generally used for this class of engine. The connexion of engine to frame and the arrangement of steam supply and exhaust pipes, are shown as well as all the valve gears. It is the cylinder with which only we have to do just now. As Corliss engines are generally of large size and long stroke the cylinder is usually made in sections *A A* (Figs. 110 and 111) is the barrel made of strong tough iron and strengthened with the bands *B B*. This barrel or cylinder casing has to take the full boiler pressure when the space between it and the liner is used, as it generally is, as a steam jacket, and for very large engines the bands *B B* are made wider and are strengthened with steel bands fitting over them.

Colin Cylinder.

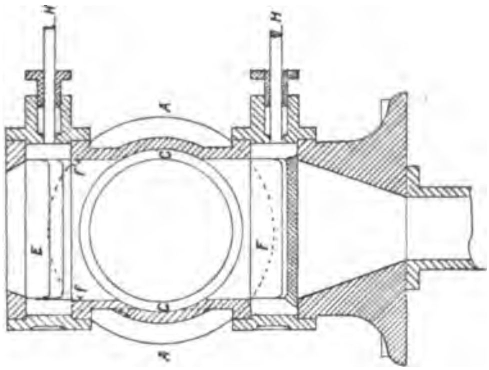


Fig. 111.

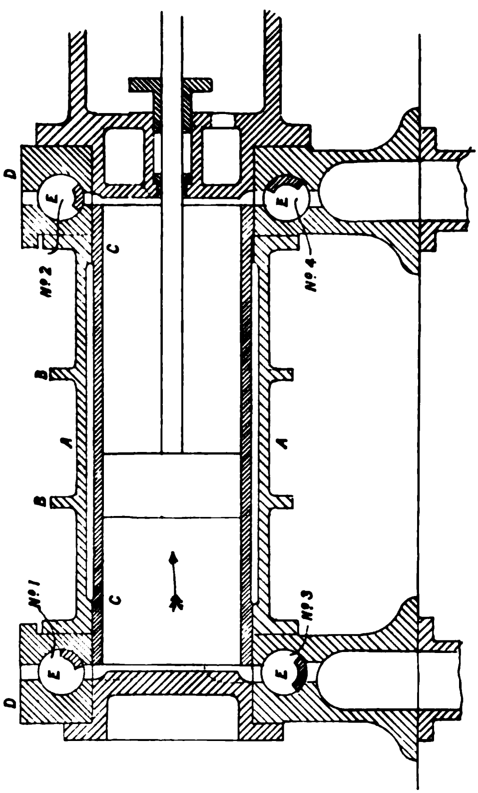


Fig. 110.

On one side of the cylinder casing a seating is cast on to carry the valve motion as shown in Fig. 125, but otherwise this is a very simple casting and is easily tooled. Within this is the liner *C C* made of strong close-grained metal and as hard as can be conveniently tooled; this is not only to prevent wear but to ensure a bright, smooth, hard surface upon which the piston can glide easily, while remaining steam-tight. The valve chests *D D* are equal in size and shape, and being bored out to fit on the projecting ends of cylinder liner, centre themselves quite truly in a line with same. In each valve chest two holes are bored out at right angles with the cylinder and these are numbered 1, 2, 3, 4 in Fig. 110. The holes extend the whole width of cylinder, as is shown in Fig. 111, which is a transverse view. Within each of the four cylindrical holes is a valve as shown, the valve fitting the cylinders at each end

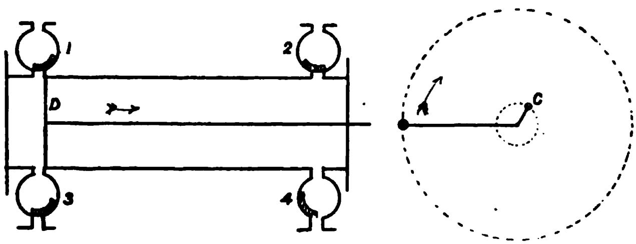


FIG. 112.

so as to keep it in place, but reduced to a section of a cylinder for the length of the ports, i.e. from *F* to *F* (Fig. 111). The two upper valves are for the steam, and the two lower ones for the exhaust. These valves are connected to spindle *H H* on one of the ends, these spindles extending through their respective stuffing-boxes and having levers on their ends worked from the valve gear, details of which we shall give later.

In Fig. 110 the valve No. 1 is shown open, No. 2 closed, No. 3 also closed, and No. 4 open. Thus the pressure of steam entering by No. 1 is driving the piston in the direction of the arrow, the exhaust steam on the other side of piston freely escaping by valve No. 4 which remains wide open for the greater part of the piston stroke.

Having now clearly seen the construction of a Corliss cylinder we will deal with one stroke of such an one, by means of outline diagrams (Figs. 112 to 115).

In Fig. 112 the piston *P* is shown just ready to take the forward stroke in the direction of the arrow, and No. 1 valve is just on the edge of the port ready to open. We will first deal with the valves as connected to the excentric *C* at an angle of 120° with crank, the same as a

Position of Valves.

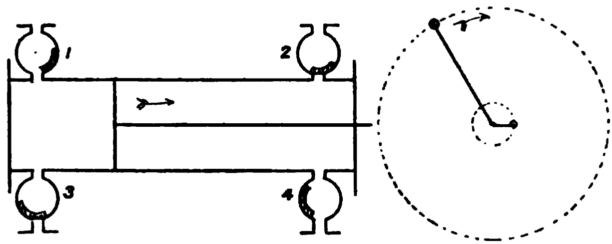


FIG. 113.

slide valve; the No. 2 valve is also connected with the same excentric, and, as seen, it just closes the port at its end of cylinder. The exhaust valve No. 3 is quite closed, and exhaust valve No. 4 is open for free passage of the exhaust from cylinder.

Now when the piston has advanced till the crank has made $\frac{1}{3}$ of a revolution and has got into the position shown in Fig. 113, No. 1 valve will be wide open and will continue open till the crank has made another third of its revolution; No. 2 will have moved to position shown, still closed; No. 3 also

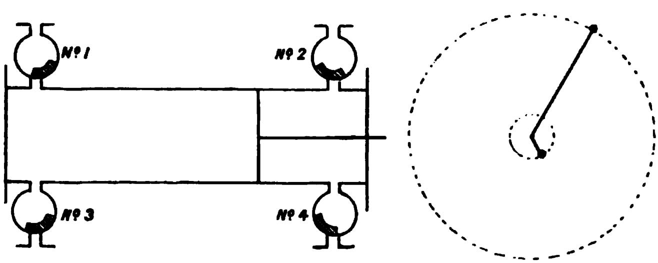


FIG. 114.

remains closed, and No. 4 is in its widest open position, and with any further motion of crank will begin to close again.

We now advance another third of the crank revolution, by which time the piston will have made $\frac{3}{4}$ of its stroke, as in Fig. 114. No. 1 has just closed, preventing any further

admission of steam. No. 2 still covers the steam port, but has got into a position where a little more travel in the same direction will open the port. No. 4 exhaust valve is on the point of closing, which it will do when the piston has advanced another $\frac{1}{8}$ of its stroke, and No. 3 will open at the same time.

In Fig. 115 we get to the last position. The crank has now

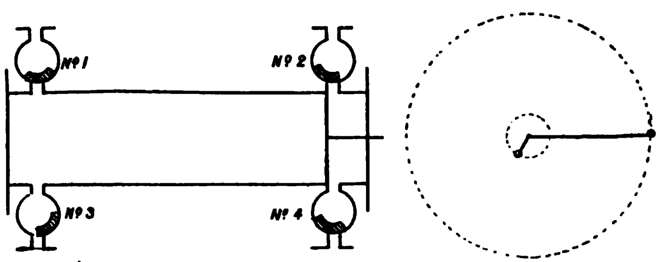


FIG. 115.

completed one stroke; No. 2 valve is just opening ready for the return stroke, No. 4 exhaust valve remains closed, No. 1 steam valve is closed, and No. 3 exhaust is now open, and all the above positions will be repeated in the return stroke.

We have with the aid of these diagrams (Figs. 112, 113, 114, and 115) shown how the four valves can be actuated from a single excentric just as a slide valve is, and so far to all intents and purposes they are slide valves with curved faces. By separating one valve, however, into its four component parts, we have been enabled to diminish the travel considerably, as it will be seen on consulting the diagram that each valve travels only a little more than double the width of its port. It will be further seen that for about half the time of working the valves are in equilibrium; thus in Fig. 115 only No. 1 valve has the pressure of the steam upon it, and it has only a small motion; No. 2 steam valve the moment it opens, remains in equilibrium until it closes again, i.e. for $\frac{3}{4}$ of the stroke; No. 4 exhaust valve has the pressure of cylinder steam only upon it, and if the engine is worked expansively the pressure diminishes on the valve as it does in the cylinder, while No. 3 exhaust will work in equilibrium until it closes again at $\frac{7}{8}$ of the stroke.

Equilibrium of Valves.

Taking into account the smaller travel of these valves and the lesser pressure upon them, there is a saving in the power

required for working them of fully 60 per cent. as against an unbalanced slide valve. This one fact alone is of great importance, for though the total power required to work any valve gear is small compared with the total power of the engine, yet there are cases when the easy manipulation of the valves is a very great consideration, as in the quick and easy reversing of engines used for winding purposes, i.e. hauling up coals or minerals from mines. These engines have often to be of great size, so much so that an auxiliary engine is used to reverse the engine, but this should never be resorted to if it can be avoided, and the use of an easily worked valve gear like the Corliss, enables a very much larger engine to be reversed by hand than is possible with slide valves.

Easy Manipulation.

Reversing Gear.

For simplicity of illustration and understanding, we have assumed the Corliss valves to be all worked from an excentric set at about 120° advance, and in the case of reversing engines that is done, i.e. one excentric for the forward and another for the backward motion, the two excentric rods being connected by a link which slides from one to the other as shown in outline in Fig. 116, where *a a* are the two excentric rods, *F* and *B* are the forward and backward centres

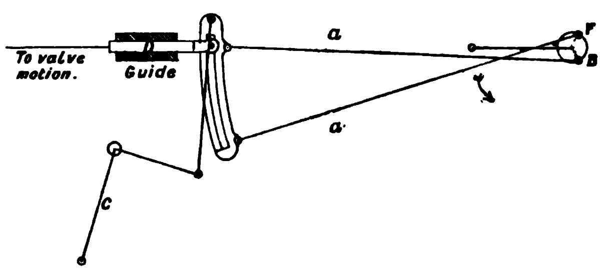


FIG. 116.

of motion, the link motion now being in the backward position. By moving the cranked lever *C* the link is lifted till the backward excentric rod works idly and the forward moves the valve, through the plunger and rod *D* which works in a fixed guide.

Reversing Corliss engines sometimes work with all the valves positively connected as we have shown them, and this is preferable for very short lifts where the direction of motion is rapidly changing, but when the mine is deep and the load

heavy the load on the engine is a very variable one, being heavy at the beginning of the journey upwards owing to weight of rope, and very light at the end. In such a case the steam valves have a cut-off or releasing motion either worked by hand or by the engine governor, and a range of cut-off from about $\frac{5}{8}$ to zero is obtained and used.

Wrist-Plate. To each of the four valves there is attached a spindle and lever, and to each lever there must be a connecting rod. In order to be able to work these four connecting rods and levers from one excentric rod a device known as the wrist-plate is adopted. A diagram of the complete connexion is shown in Fig. 117.

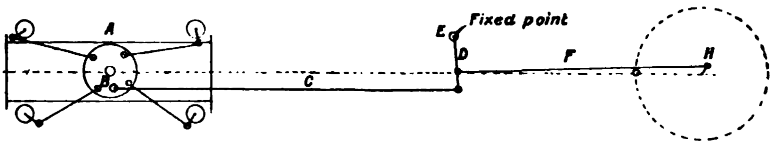


FIG. 117.

A represents the steam cylinder with its four valves, *B* is the wrist-plate which is carried on a stud pin fixed to the cylinder casing. This has to be made a firm solid support as the true working of all the parts largely depends upon it. This wrist-plate has a pin at *B* from which it receives its motion through the rod which is attached to the end of the rocking lever *D*. This rocking lever oscillates on a fixed point *E* and, through the connecting rod *F*, receives its motion from the excentric *H*.

The dotted line shows the centre line of engine, and the dotted circle the path of the crank-pin. From this outline diagram it is easy to see how the four different valves with their varying motions can all be actuated by one excentric and rod. The rocking lever *D* serves three purposes: it forms a guide and support for the excentric rod, which would be too long if it had to extend to the wrist-plate; it serves to increase the travel of rod *C*, and lastly it brings the latter rod into a horizontal line with its joint *B*, which is below the engine centre line, when the wrist-plate is worked from below the centre line.

The wrist-plate action is shown in greater detail in Figs. 118 and 119. There *A* shows the pin from the excentric

which causes the plate to oscillate between *B* and *B′*. Now if the pin which works the exhaust valve *E* be fixed as shown at *C*, i.e. at right angles with the connecting rod and wrist-plate centres, then the motion it will give to the valve *E* will

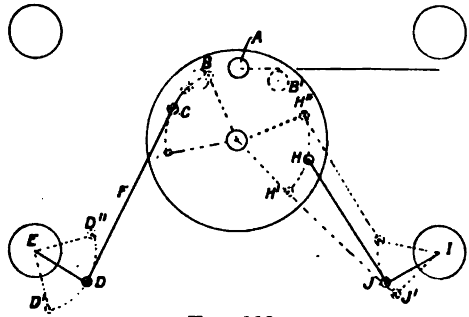


FIG. 118.

be a symmetrical or equal motion each side of the centre line as shown at *D′* and *D″*. If, on the other hand, the pin be placed at an obtuse angle with the connecting rod as shown at *H*, then while *H* will of course oscillate regularly each side of the point *H*, yet it will give a very different motion to the valve, thus while *H* is moving from *H* to *H″* the valve *I* is moved to a slightly greater distance than the valve *E* and at a more rapid rate, and also on its return till its lever takes the centre position *J*, while when the wrist-plate pin moves from *H* to *H′* it only moves the valve lever from *J* to *J′*, a very short

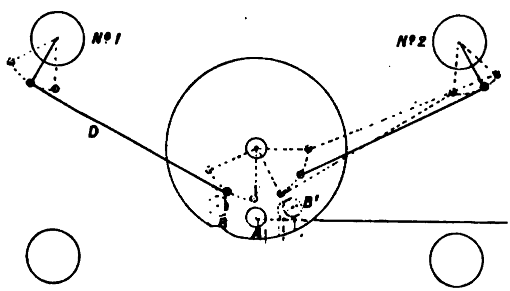


FIG. 119.

distance, and at a correspondingly lower rate of speed. By this means we can get a very rapid opening and closing of the exhaust valve, while during a great part of its stroke it remains almost stationary.

Similarly with the admission valve in Fig. 119 they can be moved more symmetrically as shown in the left-hand valve connexion, or at a varying speed as shown with the right-hand valve.

This is an incidental advantage of the wrist-plate motion, and the designer of a good Corliss gear takes advantage of the fact to get quick opening and closing of the valves. It is by no means imperative that the pins and valve levers should be placed in the positions shown; the levers may be used up as well as down, but the relative positions of lever and pins must be maintained, and these vary somewhat with each alteration in the diameter of the cylinder or length of stroke, as such change alters the position of valves with relation to the wrist-plate.

Adjustable Coupling Rods. All the coupling rods from wrist-plate to valves are made adjustable in their length, so that the admission, lead, and compression, can be readily set or altered when required. One of the simplest ways of doing this is by constructing the coupling rods in two parts as shown in Fig. 120. There *a* shows one end of the rod and

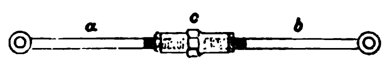


FIG. 120.

b the other; one has a right-hand thread cut upon it, and the other a left-hand thread. A coupling nut *C* with corresponding threads connects the two, and a lock nut at each end of the nut makes all firm when the adjustment is completed. It is evident that by turning the nut *C* in one direction the rod will be lengthened, and by turning it in the contrary direction it will be shortened.

Releasing Gear. One of the chief advantages of the Corliss gear is the releasing mechanism. There are several methods of effecting this, and we will consider two of the best and most widely used. In one of them the rods connecting the wrist-plate with the admission valve lever, are made in two parts, which can be automatically joined and disconnected. The method adopted is shown in Fig. 121 to a larger scale, as there is a good deal of detail in it, though it is very simple when done.

A is the part which connects to the lever of No. 1 steam valve, *B* fits on the stud on the wrist-plate. The whole apparatus takes the position occupied by the rod *D* in Fig. 119. *B* has a screwed connexion at *C* by means of which its length can be adjusted. The square block *D* has a round rod *D′*,

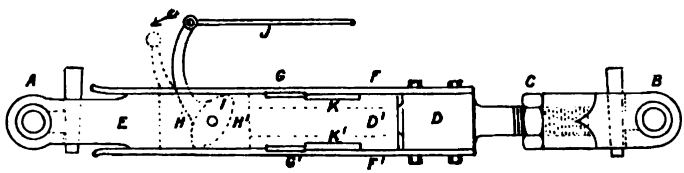


FIG. 121.

shown in dotted lines, as part of itself, this rod *D′* working freely in the bored hole in *E*. To the upper and lower sides of *D* are bolted plates made of good spring steel; these extend, as shown, nearly to the end of *E*. At *G* and *G′* hardened steel plates are let into the springs and firmly secured thereto. Sometimes these are made part of the springs. *E* consists of a rectangular steel rod with a joint at *A* which connects to the admission valve lever. At *H* to *H′* is a cavity slotted in the square rod, within which cavity the cam *I* is fitted, shown in dotted lines, and working on the joint pin as shown. A lever forming part of the cam extends through a slot in plate *G*, and is connected through the rod *J* to a hand lever on the reversing quadrant of a reversing engine, or to the governor of a driving engine.

Let into the rod *D*, and firmly secured to it, are two hardened steel plates *K K′*.

When all the parts are in position as shown in Fig. 121, the

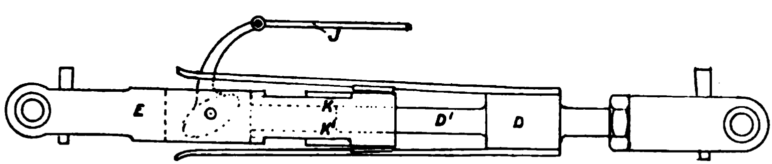


FIG. 122.

steel plates *G G′* just drop and fit into a recess in front of the plates *K K′*, thus connecting the two parts of the rod together. When so connected the rod works as if it was all in one piece, and in bringing a Corliss winding engine to rest before reversing it is always advisable to connect the rods.

The cam *I* (Fig. 121) is shown just touching the plates *G G′*. If now a little push be given to the cam lever in the direction of the arrow, the lever will tend to take the position shown in dotted lines, the ends of the cam will press against the plates *G G* and separate them; this will lift the steel plates out of their recess and disconnect the two parts of the connecting

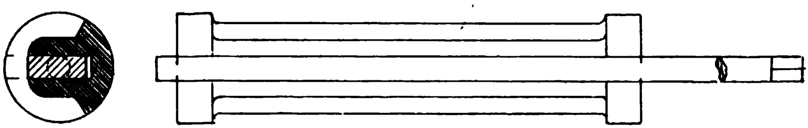


Fig. 123.

rod. As the admission valve lever has a weight or spring attached to it, always tending to keep the valve closed, the moment the connexion is broken the end *E* flies back as shown in Fig. 122 and the valve is closed, remaining closed until, in the course of the revolution, the part *B* approaches *A*, when the plates will slip together again as in Fig. 121. In flying apart, as in Fig. 122, the cam ends lie back and leave the plates free to close again, meantime *G G* travel and rub upon the smooth surface of *E*. Fig. 123 shows a detail of Corliss valve with its rod.

Having got our releasing gear to the admission valve, we are now enabled to do what we could not formerly, i.e. keep

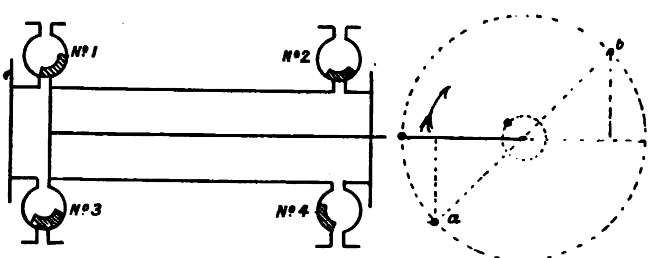


Fig. 124.

our admission valve wide open until the very moment of closing. When we had the admission excentric set at an advance of 120° we were $\frac{3}{8}$ of the stroke opening the valve and $\frac{3}{8}$ of the stroke closing it, for an admission of $\frac{3}{4}$. Now that it is closed suddenly we may make our angle of advance much less. With an angle of 45°

Position of Excentric.

the excentric will have made its full travel by the time the crank has got into the position shown in dotted lines at *a* in Fig. 124; the re-engagement will then take place, and the valve will commence to move again, first by the amount of its lap next the lead, and by the time the crank has got to its centre line, as shown in black lines in Fig. 124, the valve will have begun to open and will continue opening until the valve excentric has completed its travel and the crank has got to the position shown in dotted lines at *b*. It is essential that the release and closing of the valve should take place before this point is reached, or the admission valve would remain open on the return stroke, but the release of Corliss valves generally takes place in the first half of the piston stroke.

Double Wrist-Plate. It is evident that while such a position of the excentric is of great advantage to the admission valve, it would not do at all for the exhaust valve, so the practice is adopted of using two excentrics, one for the admission and one for the exhaust, and having two corresponding wrist-plates on the same stud pin, one behind the other. The exhaust valves can thus be set quite independently of the admission and have just the lead and compression given to them that is required. The same independent adjustments can be made with the admission.

A very good arrangement of Corliss valve is shown in Fig. 125. This illustration is supplied through the courtesy of Messrs. Hick Hargreaves & Co., of Bolton, one of the most experienced and celebrated makers in the world. The figure shows the arrangement of the Corliss motion very clearly. The exhaust valves are worked from the back wrist-plate, which receives its motion from the rocking lever and excentric nearest the engine frame. The front wrist-plate works the admission valves and receives its motion from the rocking lever suspended below the governor stand. The exhaust valve levers are simple and need no description, but the admission levers will be seen to have cranked arms, the rods from the short arms being connected to pistons within the horizontal cylinder seen above the wrist-plate.

Compressing Cylinder. Strong springs within the cylinder close the valves as soon as they are released, and the piston in rushing back compresses the air behind them. This makes a cushion at the end of the

Fig. 125.

stroke, this cushion destroying the shock and preventing the jar which would otherwise take place.

Governor Connexion. The connexion of the releasing lever to the governor is also seen, but this we shall deal with later under its own head. Altogether the illustration shows a very good example of a high-class Corliss gear.

There are many varieties of Corliss releasing gear, but they may all be divided into two types: one which is called the "scissors" motion, which we have described, and the other that known as the "crab-claw" motion. This is shown in

Scissors Motion, Crab-Claw Motion. Fig. 126. No. 1 shows a lever which is firmly secured to the No. 1 admission valve spindle; as it is exposed to considerable strain and shock it is generally fixed on to a square on the valve spindle. At *a* is a hard steel plate, and at *b* a joint and rod,

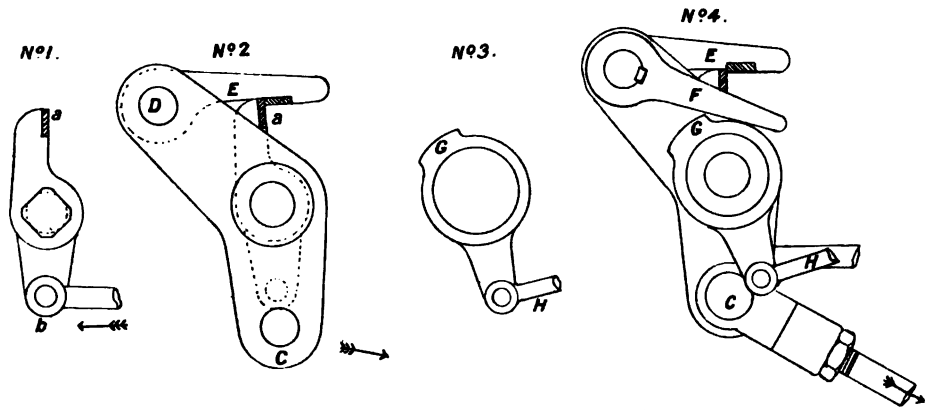


FIG. 126.

connected to a powerful spring which forces the valve to close by pushing the lever in the direction of the arrow.

On the same valve spindle, but moving freely upon it, is the bent lever No. 2. This is connected at *C* to the wrist-plate by an adjustable rod, the joint of which is shown in No. 4, and at *D* it carries the catch lever *E*. This lever has also a hardened steel plate let into it; the two plates being shown in this and the other figures by shaded lines. The catch lever is fixed in such a position that it easily engages with the valve lever No. 1 shown in dotted lines in No. 2. When No. 2 is

pulled (in the direction of the arrow) from the wrist-plate, the lever *E* will compel the end of the valve lever at *A* to move in the same direction at *D* and thus open the valve. In order to lift the lever *E* at any desired point and thus release the valve lever and close the valve, there is another lever keyed tightly on the same spindle as *E* but at a different angle; this is shown in No. 4 at *F*. It is evident that if this is pushed up *E* will go with it, and the plates will be disengaged. This is accomplished by the cam and lever No. 3, *G* being the cam and *H* the connexion to the engine driver's hand or to the engine governor. By means of the rod *H* the cam which encircles the boss of No. 2 lever can be moved forwards or backwards.

In No. 4 all the parts are shown in position together, and it will be seen that as the point *C* is pulled by the wrist-plate in the direction of the arrow, the cam *G* remaining stationary, *F* will come into contact with the cam and be lifted, disengaging *E* which is keyed firmly to the same spindle as *F*.

Like the "scissors" motion, the "crab-claw" has a good many parts in its composition, but these are easily understood and the action is satisfactory. By moving the cam rod *H* by hand, as is frequently done in winding engines, the engine can be controlled by the driver through the cut-off gear as well as by the reversing gear and stop valve. In drawing from very deep mines it is a regular practice to start with a full admission, gradually advancing the point of cut-off as the load comes to the top.

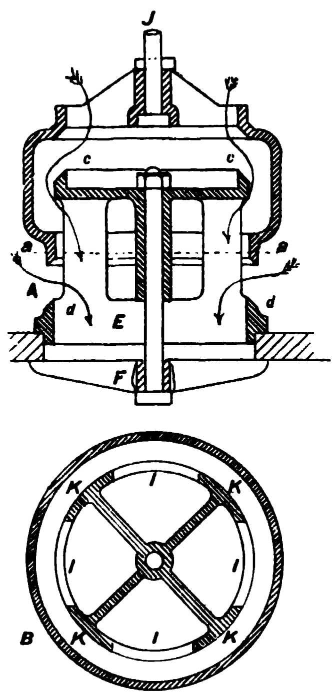
Notwithstanding all its many advantages, it cannot be denied that the Corliss gear is somewhat complicated, and unless the engine driver be a little above the average in intelligence, it may get out of order. Many years of successful working, however, has established it as one of the best that has been introduced.

Equilibrium Drop Valve.

We shall now deal with a quite different style of valve and valve motion, viz. the equilibrium drop valve. This, though one of the oldest, has only during the last twenty-five years come largely into use, Messrs. Sulzer & Co. on the continent and Messrs. Robey & Co. in England being principally instrumental in its introduction, but above all, the much higher pressures of steam which are now used, as compared with thirty years ago, have almost

compelled the use of this form of valve. The Corliss valve gear works at high pressure with but very little friction, the equilibrium valve with practically none.

Cornish Valve. The equilibrium valve is a very old one, having been used in some of the earliest Cornish pumping engines, and it was long known as the Cornish valve; its original form is shown in Fig. 127, *A* being a vertical section, and *B* a horizontal section of such a valve. Fig. 127 shows the valve in its open position, the arrows showing the path of the steam. When open the valve is entirely surrounded by steam and is in perfect equilibrium, and when closed all is in equilibrium with the exception of the small area of one of the seats.



Section on Line A.A.

FIG. 127.

The lower seat *d d* has to be slightly larger than the upper seat *c c*, as the part *a a* has to pass over the upper valve seat. The little difference in pressure is very small, and is not more than enough to secure the tightness of the valve by the extra pressure of steam holding it on to its seat. The very slightest movement of the valve from its seat admits steam between the faces, and the valve is then in equilibrium through the whole of its motion, and until it is settled upon its seat again. The valve seat *E* is secured to its seat by the crossbar and bolt passing through the valve. This part has a number of openings in its side shown at *j*, *j*, *j*, *j* in the plan, the bars ***k***, ***k***, ***k***, ***k*** connect the upper and lower seats together, and also form guides for enabling the valve always to fall truly on its seat. The steam passes through the openings and the valve itself is attached to the rod *J* going through a stuffing-box above, and from this rod, which is in tension, it receives its motion.

A modified form of equilibrium valve is shown in Fig. 128, and it is the form which is generally used in modern steam engines. Here the valve seat *A* instead of standing above the partition plate, is suspended from it, the upper seat *B B* by which it is fixed being made very substantial and strong, as it is most important that it should not only be firm but also absolutely true.

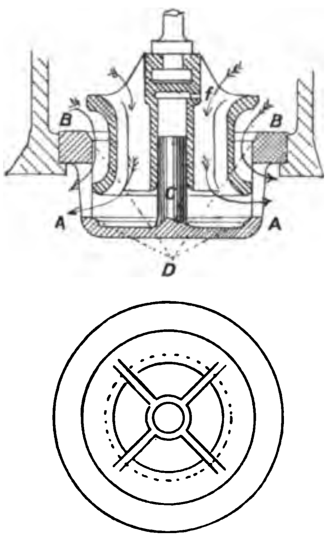


Fig. 128.

The upper and lower seats are connected by strong bars, in the spaces between which the steam passes, the path of the steam being indicated by the arrows.

The valve is guided by a strong cylindrical bar *C* rising from the centre of valve seat, the boss of the valve is bored to fit this truly but easily, the small hole at *f* being made to allow of free inlet and exit of the steam so as to avoid cushioning.

Angle of Valve Seats. There is one matter which is of more importance than seems at first sight, viz. the angle of the valve seats. If these are made at 45° as was formerly the practice, it is very difficult to keep them steam-tight, and they require frequent grinding for that purpose, but an angle has been found which gives entire satisfaction in this respect, valves so made remaining steam-tight for many years. To secure this angle let a point *D* be fixed ¼ of the diameter of the valve below the lower seat; from this point draw lines as shown to the seats *A A* and *B B*, and these lines will give the required angle of the seats whatever be the diameter and depth of the valve. It is important also that both the valve and its seatings be made of the same metal, so that they may expand equally with equal temperatures. The best material for the purpose is phosphor bronze, though some makers use cast-iron, which may be permissible for engines of large size and few revolutions per minute, but for speeds of 150 openings per minute, up to which speed the author has used large

valves of this kind, phosphor bronze is the most satisfactory material.

It will be noticed that the valve is double seated, hence a very small lift gives a quick and wide opening. Thus a valve of only 6 inches diameter, being over 18 inches in circumference with its two seats, is equal to a slide valve of over 3 feet length of port. A usual practice is to make the valve $\frac{1}{3}$ the

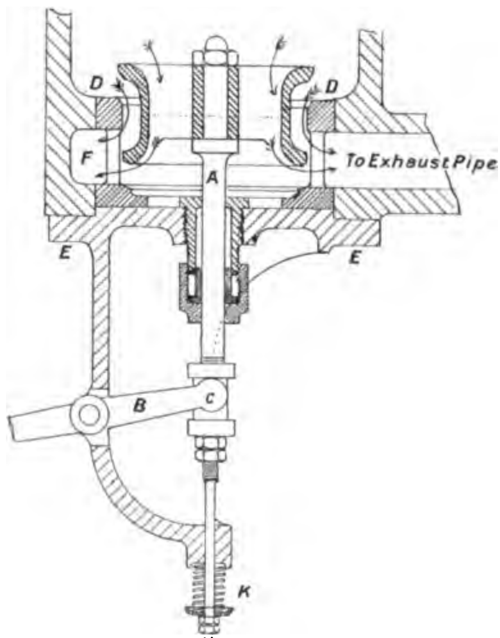


FIG. 129.

diameter of the cylinder, but this of course depends upon the pressure of the steam and the speed at which the piston has to work.

Drop Exhaust Valves.

The same kind of valve is largely used for the exhaust as well as admission. Fig. 129 shows an arrangement in use by some makers. Here the valve instead of being suspended is supported by its rod when off its seat, the rod *A* passing through a long stuffing-box which serves as guide, the lever *B* being thrust down at its outer end in order to raise the point *C*, which lifts the valve spindle. The valve seating is kept in position by the flange

of the bracket *E E*. The part *D D* of the seating has to make a steam-tight joint at *D D*; fortunately the expansion by heat is greater by nearly double the amount in copper and its alloys, than it is in cast iron, and though the amount is

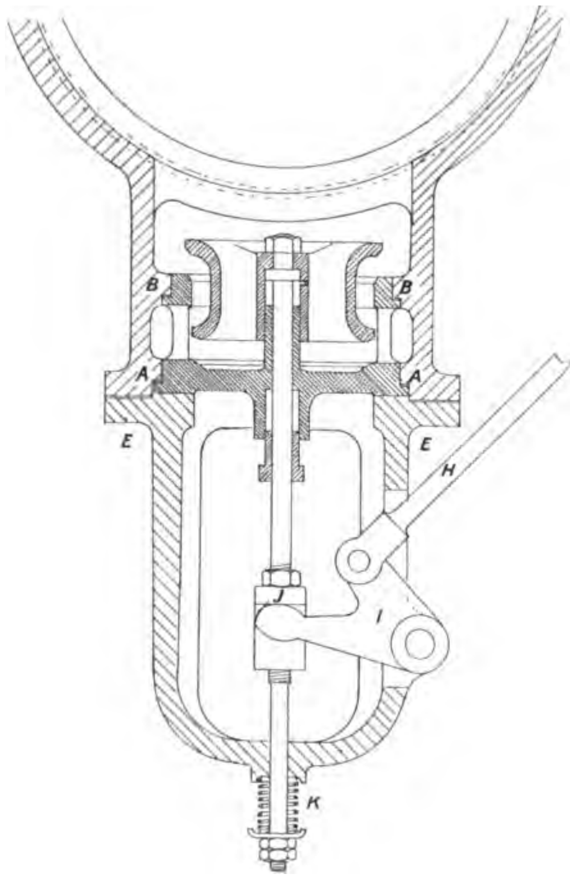


FIG. 130.

small, viz. ·001 in iron and ·002 in gun-metal, yet this fact tends to keep the joint *D* tight under steam. The arrows show the path of the steam, there being a chamber *F* all round the valve and a passage at one side to the exhaust pipe.

What in the author's opinion, is a much better method of forming an equilibrium exhaust valve is shown in Fig. 130.

Instead of a plain joint as in Fig. 129, in this method there

is a recess formed at *A A* and *B B* and a corresponding flange on the valve seats. The seating can then be made a fairly easy fit in its socket, steam tightness being secured by a thin ring of comparatively soft metal or other suitable jointing material between the flanges and the seat, as shown by the parallel lines. Where the bracket *E* is bolted up into its place, it will make the joints steam-tight, and when removed, the valve seat can be more easily withdrawn. The hollow vertical peg *F* is also a much more satisfactory method of guiding the valve on to its seat. With Fig. 130 there may be, and there is, a greater weight of a more expensive metal, but the advantage of such an arrangement fully justifies the extra cost. In this arrangement the valve is lifted by the lever *I* being drawn up by the excentric rod *H*, the cam end of the lever presses on the shoulder of the nut *J* in order to lift; *J* can be adjusted vertically, being screwed upon the valve spindle, and being locked in position by the lock nut above it.

There are no shoulders or collars below the cam, as it is necessary to give some freedom to the lever to move after the valve has got to its seat.

This arrangement has the incidental advantage that the exhaust excentric may have a much greater travel than the valve needs, thus its opening and closing may be made rapidly in a part of the excentric travel, the remaining part being quite free, until the time for opening comes again.

Disadvantage of Exhaust Drop Valve.

A coiled spring on the outside of bracket at *K* tends to keep the valve quietly on its seat.

One of the disadvantages of this form of valve is its difficulty of access. It occupies, as is seen from the same section of cylinder in Fig. 130, a considerable space under the cylinder extending some distance below the floor level as is seen in the transverse section (Fig. 131). Not only is the exhaust valve placed in a somewhat inaccessible position, but its placement interferes with the construction of the foundations, increasing their cost, and tending to lessen their stability. The stability can of course be increased in other ways, among which is the provision of a massive cast-iron foundation plate, shown partly in section at *A A*, and the making of a chamber under the cylinder in order to give easier access to the exhaust valve gear. But when all this

has been done, the old proverb that "out of sight is out of mind" largely holds good. Little attentions which may be

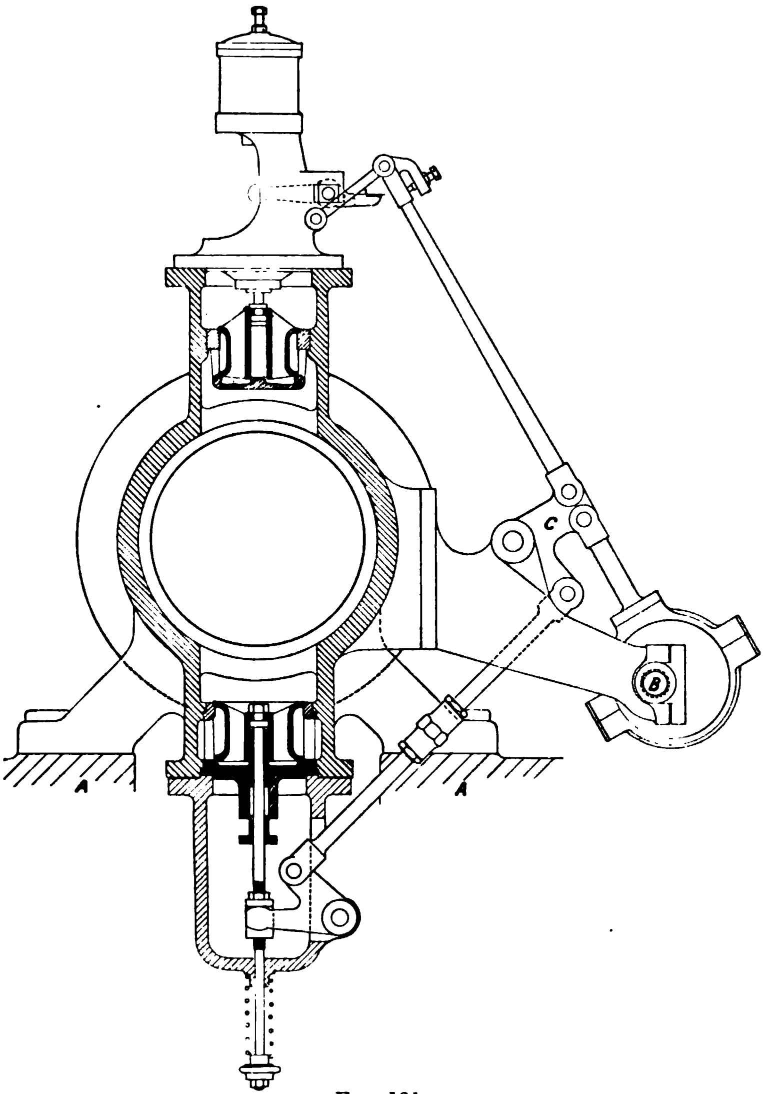


FIG. 131.

needed, and that can, and would, be given were all the parts easy of access, often get delayed and sometimes altogether neglected, to the detriment of the good and economical working

of the engine. Nevertheless, this exhaust valve and gear is good and is largely used by some of the best makers, especially on the Continent.

Excentric Shaft. In this (Fig. 131), a clear view is given of the general disposition of the admission and exhaust equilibrium drop valves, and the method of working them by excentrics on a supplementary shaft *B*, running parallel with the centre line of the engine and containing all the excentrics. The cranked lever *C* shows a method by which the exhaust as well as the admission valve may be worked by one excentric.

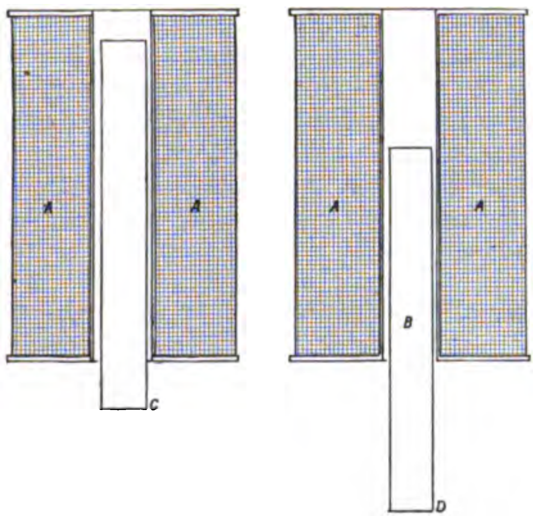


Fig. 132.

This is an ingenious arrangement and valuable when such valve gear is made reversible, but for ordinary driving engines it is preferable to have a separate excentric for each valve.

We shall next deal with the "Richardson" valve gear, in which the desiderata for convenient and economical working have, in the opinion of the author, been best secured.[1]

Electric Control. At that time, 1881 to 1887, the author had been experimenting for some time with the electric control of large engines, and was in search of a form of expansion gear which could be controlled within wide extremes without putting any perceptible strain upon the governing

[1] Patent No. 5,744, 1887.

gear, which was a sensitive electric solenoid (Fig. 132). The solenoid consists of a great length of well-insulated wire wound upon the copper reel *A*. This coil becomes magnetic when an electric current is sent through it, so much so that the heavy soft iron core *B* remains poised within it, as shown, without visible means of support. If properly proportioned, a 2 per cent. variation in the strength of the electric current will vary the position of the core from *C* to *D*, with corresponding smaller changes for lesser variations in the current. Nothing can be more delicate than this, or more perfectly free from any mechanical friction.

The Solenoid.

Not finding in any existing valve gear the extreme sensitiveness required, the author designed and adopted that shown in Figs. 135 and 136, of which the former is a transverse section, and the latter a longitudinal section through a steam cylinder and valve chests fitted with such a gear.

The admission valves *A* are described on p. 140. They are lifted by the lever *B* by depressing the outer end *C* by the tripper *D*, which is actuated by an excentric and rod, or by a rocking lever. It must be seen that the longer time that the tripper *D* remains in contact with the lever *C*, the longer time the valve *A* will remain open; and it is by varying the duration of this contact, that the varying points of release have been secured. Hitherto this had been done by controlling the position of the tripper, as shown in Fig. 133. In this case, the valve lever *E* (Fig. 133) is cranked as shown at *R*, and is actuated by the tripper *I* which has an arm *S* at right angles with it. When the excentric rod *C* is pulled in the direction of the arrow, it carries the tripper with its crank arm with it, and as soon as the arm *S* comes in contact with the stud *T*, it is arrested, and any further motion of the tripper rod *G* would tip up this arm and thus release the tripper *I*.

Trip Gear.

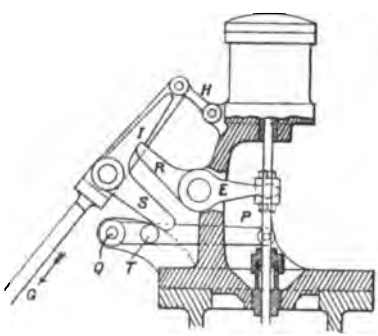


Fig. 133.

The stud *T* is curved on the lever *P*, one end of which is jointed at *Q* and the other end connected with the engine

governor in any convenient way. There are many varieties of this method of releasing drop valves, and they work well with ordinary centrifugal governors, but the mechanical strain between the stud T and the lever S is sufficient to cause large fluctuation in the position of a solenoid, and even to cause a centrifugal governor to slightly change its position, if at all sensitive.

Excentric Cut-off Gear. Another arrangement which was made and tested is shown in Fig. 134. Here the tripper I is carried on the end of the excentric rod G and has a simple up and down motion. The tripper being jointed falls back

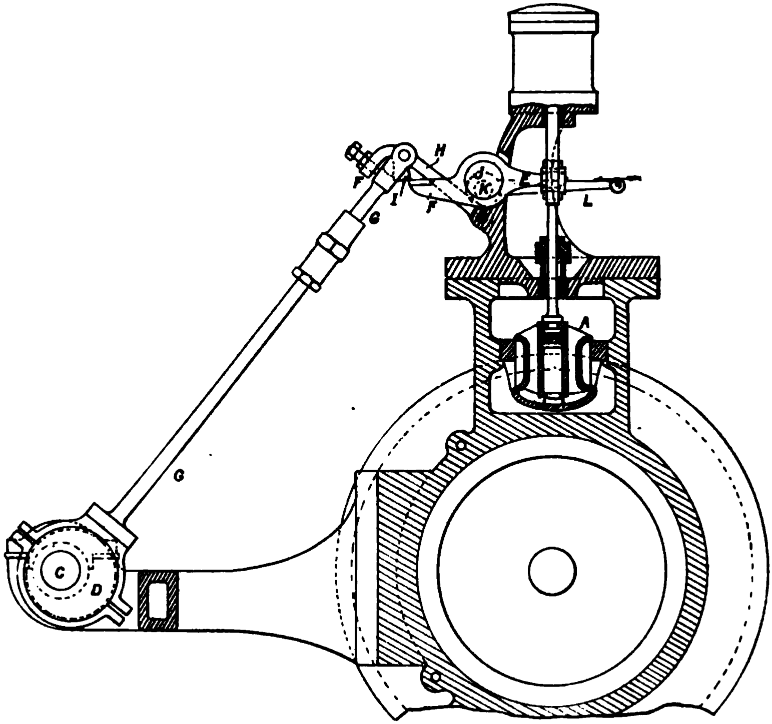


FIG. 134.

as it goes up, and as soon as it is clear of the valve lever, falls forward on to it, at its outer end F.

The valve lever has its fulcrum upon the bearing J, which as is shown is excentric to the rod K upon which it moves. Upon K is a lever L connected to the governor. It will be seen that

as L is raised or lowered the centre of the excentric J is moved to the right or to the left, giving a longer or shorter time of contact between the tripper and the end F of the valve lever E. As a simple releasing apparatus this worked beautifully, and were it always in its central position as shown, it would leave little to be desired; but in working, the central point of the excentric is not always vertically over the centre of rod, and the strain of the trippers just at the moment of opening the valve, is sufficient to move the excentric and thus elevate, or depress, the governor lever, as the excentric inclines to the right or to the left of the centre line.

This strain is much more than seems at first sight. In describing the equilibrium valve we called attention to the fact that it was not in equilibrium when closed, but became so at the moment of opening. The unbalanced seat of a valve 6 inches in diameter is not less than 2 square inches in area, and with a pressure of 150 lb. to the square inch the load to be lifted is 300 lb. and there is also the pressure on the tripper in order to lift the valve; the sum of the two is 600 lb. pressure on the small excentric, as it is situated between the two and bears both loads, just as the centre of a scale beam has a load on it equal to that on both of the scales. In order to eliminate this disturbance and take practically all load from the governor, the author designed and, after trial, finally adopted the method shown in Figs. 135 and 136.

Here (Fig. 135) A is the valve lifted by the lever B by means of the tripper D pressing on the outer end C of valve lever. The general arrangement is best seen in Fig. 136, which gives a horizontal section through cylinder. There, D is the valve chamber having the valve A A at each end. The drawing shows the valve arrangement so clearly that any further description is unnecessary; the only points which may need explanation in the drawing are the dash-pots E E. The valve spindles are seen to be extended upward through a well-fitting bush, and are connected to pistons F F, which are also a good though easy fit, i.e. they are made air-tight by a number of grooves made in the piston.

When the valve is closed, the true bottom of the piston should not be more than $\frac{1}{32}$ of an inch from the bottom of the cylinder, and may be even less than this; on the valve and therefore the piston, rising, air is admitted below it through

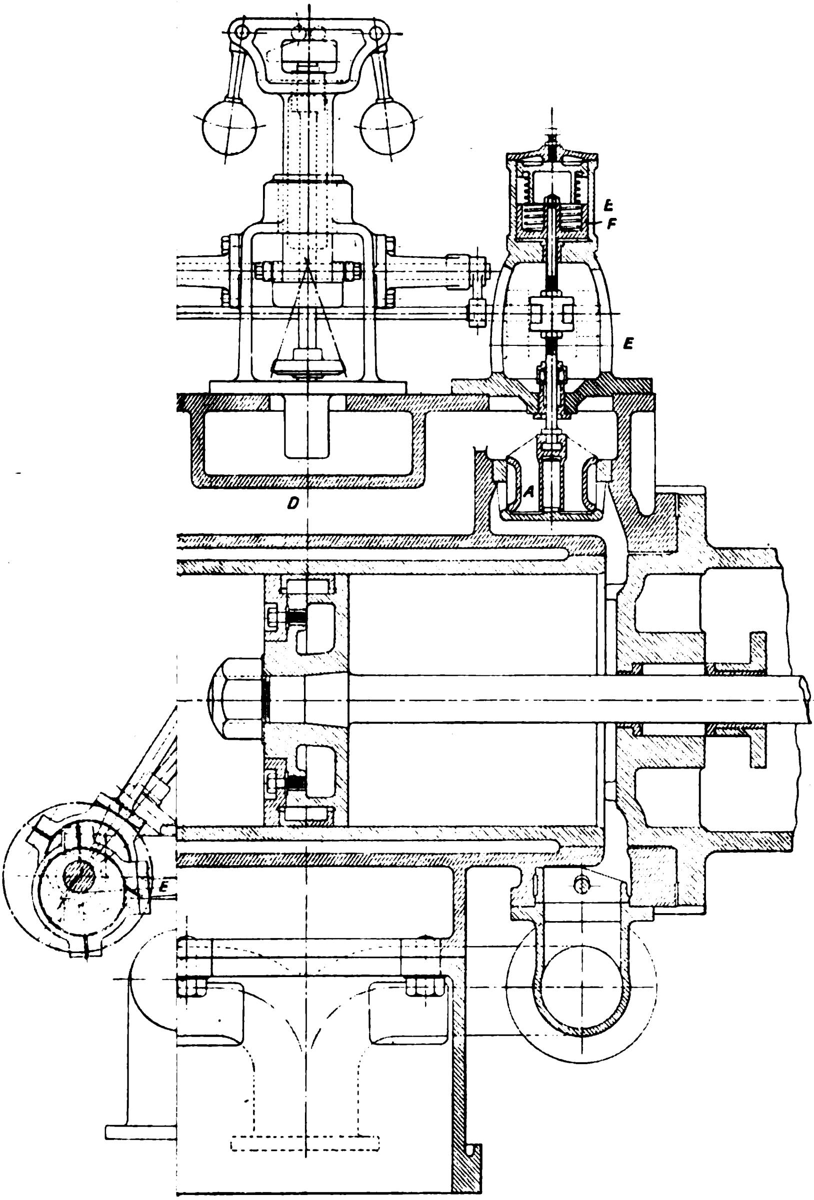


FIG. 136.

[To face page 148.

a self-acting valve, which closes (and thus imprisons the air) the moment the upward stroke is finished. When the valve is released this imprisoned air becomes a spring cushion between piston and cylinder bottom, and as it rapidly rises in pressure as the piston falls, it entirely prevents any shock or noise due to the closing of the valve. A small regulating screw is fixed on the air valve, by adjusting which exactly the right quantity of air can be permitted to escape and the valve allowed to close quickly and certainly, but without any jar.

Cushion Cylinder.

That the exact method of getting a releasing movement in perfect equilibrium (so far as the governor is concerned), can be more clearly seen, we have dissected the parts as we did with the Corliss gear.

Fig. 137 shows the method adopted for moving the centre of the valve lever, *a* is the bar on which the valve lever works;

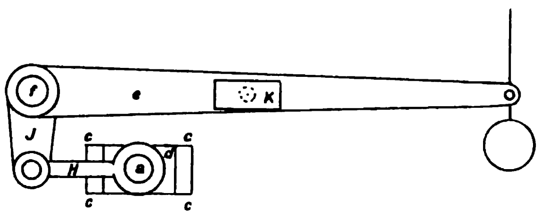


Fig. 137.

this bar goes through both valve levers and thus extends the whole length of the valve motion. The valve lever *b* is shown in position upon it in Fig. 138. *c*, *c*, *c*, *c* (Fig. 137) is a rectangular and perfectly horizontal slot in the standard carrying the dash pots and levers. Within the slot there are for each valve lever two gun-metal slide blocks *d*, free to move easily to the right and left, though only for a very small distance as shown; *e* is the governor lever with its fulcrum at *f*, and *J* is a short lever keyed on to the same rod as *e*, and thus moving with it. *H* is a small connecting link between the end of lever *J* and the rod *a*. *K* is a sleeve resting on the collar of the governor slide (not shown); the end of lever *e* is prolonged beyond the governor and carries a small weight to keep it always tightly resting on the collar of governor, as there is only one collar to the slide. It will now be seen that as *K* rises or falls, so through *J* and *H*, the slide block *d* is moved to

the right or left. In Fig. 138 the valve lever *b* is shown in position, as well as the excentric rod *L* and the tripper *N*. The end of the excentric rod *L* is connected to a pair of rocking

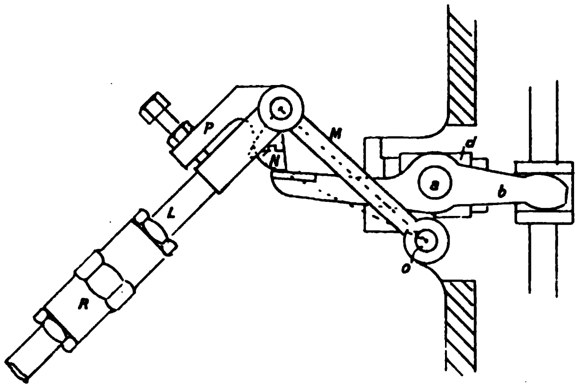


Fig. 138.

rods *M* pivoted at *O*. As the rod *I* goes upward, the tripper *N* leans against it, as shown in Fig. 139, and falls into position on the end of lever *b* as soon as it clears it. In Fig. 138 the

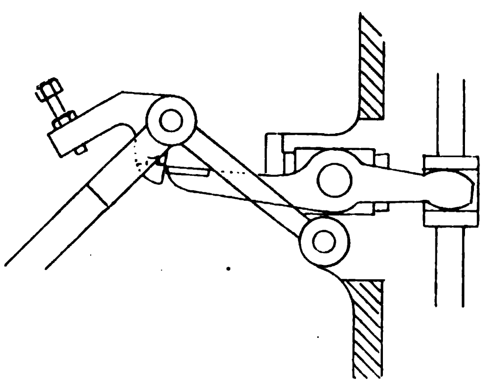


Fig. 139.

rod *L* has commenced its downward movement and is on the point of depressing the valve lever; as it does so, the connecting link *M* whose end works in an arc, carries the tripper further out until it clears the end of lever; then the lever springs back, the valve closes, and the tripper continues its downward stroke quite free. Were

Slide Block Trip.

the point *a* fixed, this release would always take place at one point, but owing to the movement of the slide block *d* this point of release is varied; as the block moves to the right the tripper leaves the end of lever earlier, and as it moves to the left it remains in contact later. The set screw and lock nut on the tripper crank *P* is for the sake of setting the front and back valves to cut-off at the same point in the stroke, and the right and left-hand adjusting nut *R* on the valve rod *L* enables the lead of the valve to be set to a nicety even when the engine is running.[1]

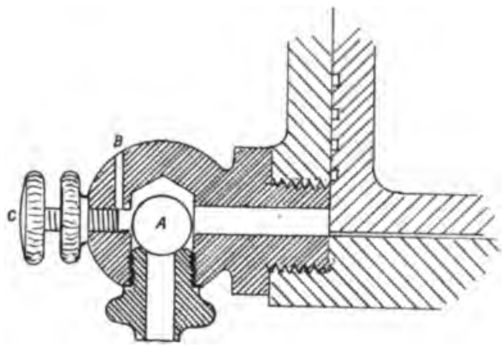


FIG. 140.

Of course with this gear there is just the same load of 600 lb. on the lever *b* at the moment of opening the valve, but the pressure being directly downward and being resisted by the surface on which the blocks slide at right angles to the pressure, there is not only not the slightest tendency to cause any

[1] The author takes this opportunity of expressing his high appreciation of the skill and ingenuity of Mr. Bartholomew Rowland, who was employed by him as draughtsman to make the drawings for the patent specification and to get out the details of the invention. The two designs (Figs. 133 and 134) were Mr. Rowland's suggestions entirely. and the author had great pleasure in including his name in the Patent.

The excentric motion (Fig. 134) for moving the centre of the levers is the simplest conceivable, and though the motion is not perfectly true, yet on trial it gave good results, and Fig. 133 is quite equal to any of the releasing gears which move the trippers. Figs. 135 and 136, which was the author's invention, was after trial found so much better in working that it was finally adopted and is in use in many thousands of engines.

The careful working out of the details was, however, principally in Mr. Rowland's hands, and contributed much to the success of the invention, which quickly enhanced the commercial success and engineering reputation of the makers, Robey & Co., of Lincoln.

movement of the governor lever *e*, but as a matter of fact just at that moment of severe strain, the strain itself holds everything tight. So little strain is there upon the governing lever that it may be suspended upon a silken thread or hung (as we shall see later) upon the cores of an electric solenoid without disturbing its position.

Delicacy of Slide Trip.

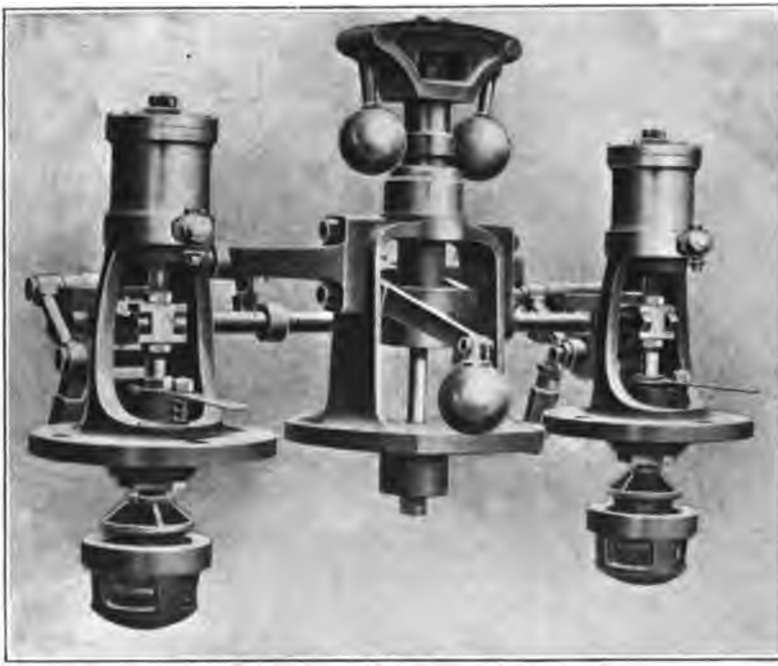


Fig. 141.

The little air valve for cushioning the dash-pot piston is shown full size for a 6-inch valve in Fig. 140, as the dash-pot piston rises, air is drawn in, the ball valve *A* rising to admit it, and when piston falls the imprisoned air has to escape by the small hole *B*, the set screw *C* covering as much of the hole as enables it to give the right cushion.

Air Valve for Dash Pot.

In Fig. 135 a line is seen above the ball at the end of governor lever *e*. As there is no upper collar to the governor sleeve the lever may be lifted when the engine is running, and this is the quickest way to stop the engine as steam is instantaneously

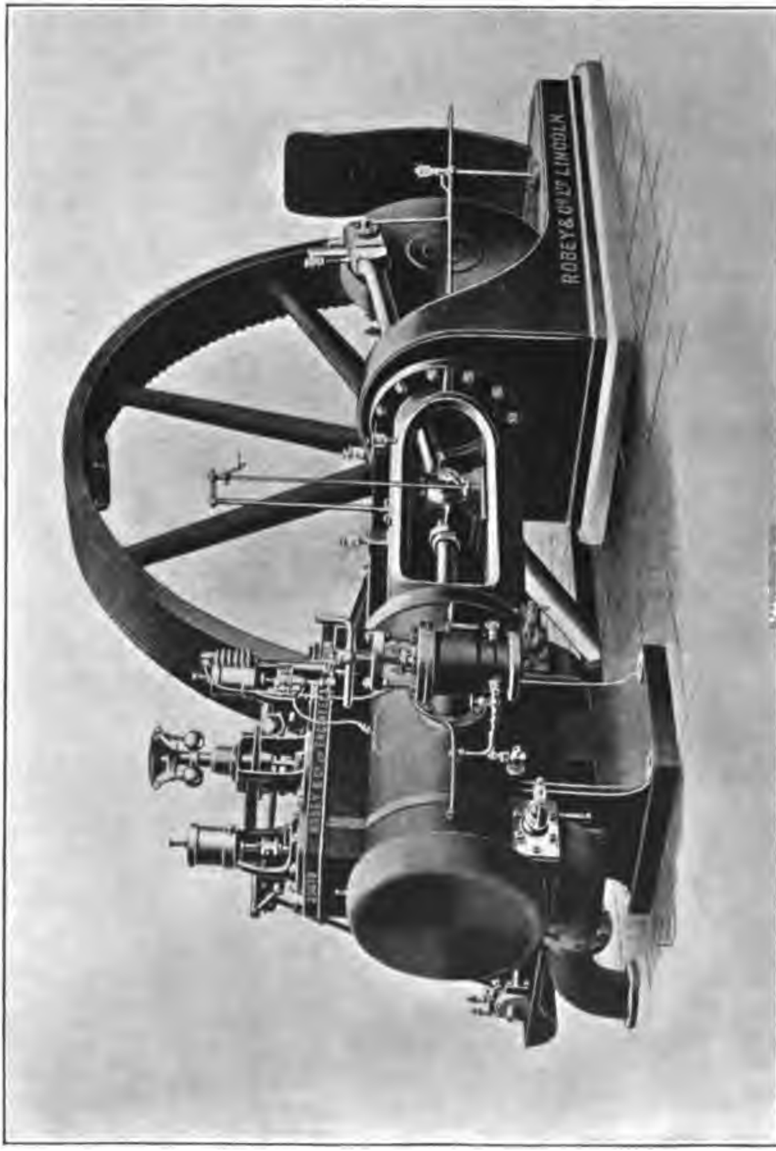


Fig. 142.

cut off from the cylinder. The cord may be carried any distance in a mill or factory, and in case of danger or accident the engine can be stopped from any distance where the cord extends.

Fig. 141 shows a general perspective view of a complete set of Richardson valve gear detached from the cylinder to which it belongs, and Fig. 142 gives a perspective view of one of the most recent designs of horizontal single cylinder steam engines fitted with it.

CHAPTER IX

Exhaust Valves and Valve Driving Gear.

In the previous chapters we have dealt chiefly with the admission of steam. The exit of steam from the cylinder, or exhaust, as it is called, is almost, if not quite, of equal importance, as without a free and uninterrupted exit there would be back pressure and diminution of power. We have already pointed out on p. 104 the advantages of ordinary slide valves. Long ago when "lap" was first put on slide valves on the outer or steam admission edge, it was also put inside on the exhaust edge so as to prevent the early release of the steam, and though the evils of inside lap have been demonstrated again and again, yet this unwise practice is frequently resorted to, with the object of getting a little more out of the steam; but, so far as the author's experience goes, always with bad results.

Exhaust Valves.

Inside Lap.

Some time ago he was called in to investigate the lack of power in a large slide valve engine, and, on taking a diagram, which had not been done before, the evil was at once apparent.

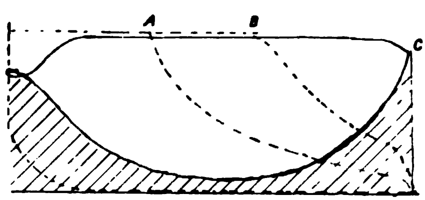


Fig. 143.

The diagram was like Fig. 143 (consisting of the unshaded part), showing inside lap and a late admission, and an examination of the slide valve confirmed this opinion. On removing the inside lap, and setting the valve properly, the diagram became

like the dotted lines, and the change naturally resulted in both a considerable saving in steam and increase of power. All the shaded part of diagram being back pressure, the removal of this increased the power of engine, and the earlier cut-off at *A* economized the steam, while the late cut-off at *B* gave a great margin of power.

There are doubtless many thousands of steam engines working even at this day with as wasteful consumption of steam as that shown in above diagram.

The principal things to be considered in designing an exhaust valve are—quick opening and closing, a clear passage for the exhaust steam (at a speed not exceeding 100 feet per second, or 6,000 feet per minute), small clearances, easy working, and accessibility.

A properly proportioned slide valve gives us all these with the exception of easy working. It is convenient to work, but, as we have already shown, it is heavy to work in large sizes, owing to the friction.

The Corliss exhaust valve is exceedingly good, simple to work, very accessible, and, when properly made, leaves little clearance. To make the clearance as small as possible while leaving a free passage for the exhaust, the valve itself should be made bulky, as shown in section in Fig. 144, so that all the space in its working cylinder can be filled that is not needed for the passage of the steam. There is but little friction with this valve, as all the time it is open (which is half its stroke) it is in equilibrium; and during the time it is closed it has only the cylinder pressure upon it instead of the boiler pressure, as with the slide valve. This valve has been largely used in connexion with drop valve admission, but of late years the equilibrium valve has also been used for exhaust. So far as ease of working is concerned and freedom of exhaust, the equilibrium valve is certainly one of the best, and a good example of its use is shown in Fig. 131, p. 144. About the only objection to its use is its inaccessibility and the way it interferes with the foundations.

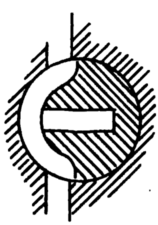


Fig. 144.

Gridiron Valve. In the author's opinion, the multi-ported or Gridiron slide exhaust valve used by him in connexion with the drop equilibrium admission valve, makes the arrangement

which combines the greatest number of advantages, with the fewest defects. This is shown very clearly in Fig. 135.

The valve, as there shown, is situated immediately below the cylinder, so that the latter always remains drained during the time the valve is open, which is nearly the whole of the return stroke. Should there be any little water left in the cylinder at the moment of closing, say at $\frac{7}{8}$ of the stroke, this can escape by the relief valve shown at *H*, but the cylinder being so well drained automatically, this very rarely comes into play. Ordinary cylinder relief cocks are seldom needed though provided. The valve is situated at the least possible distance from the cylinder, and it fills the whole of the chamber in which it works, less the amount required for its travel. The travel itself is very small, a valve with four ports having only $\frac{1}{4}$ the travel that it would need if there were only one. It is easily and conveniently worked by the excentric and rod *E*, and, as is seen, it is of very ready access for examination and adjustment, and can be removed if required in a few minutes by removing the cover *F* and unscrewing the valve rod *C*.

Like the Corliss valve, it is in equilibrium when open, and has only the cylinder pressure upon it when closed. The clearance is very small and it is because of this fact and that water is incompressible that the relief valve *H* is used, as even a very small quantity of water might do much damage were there no exit for it. It is, however, very rarely that the valve comes into play.

As shown in Fig. 135, the valve is just opening, and from the position of its excentric it will be seen that it is just at its most rapid motion. If set with $\frac{1}{8}$ advance the exhaust will be practically wide open by the time the piston has got to $\frac{1}{8}$ of its return stroke, continuing wide or nearly wide open till it has made $\frac{3}{4}$ of its stroke, and closing for the compression at $\frac{7}{8}$. Another advantage the valve has is the very free discharge for the exhaust steam after it has passed the valve; there are no tortuous passages to traverse or sharp corners to turn, but the steam at once passes into the spacious chamber *J*, from whence it is conducted to the atmosphere, the condenser, or to the second cylinder of a compound engine.

From the one shown in Figs. 145 and 146, it is seen how little steam there is above these valves, and how ample is the space below them.

Before considering the further details it would be well to examine Figs. 147, 148 and 149. These figures are from a working drawing, showing respectively the side elevation, end elevation and plan of a horizontal coupled compound

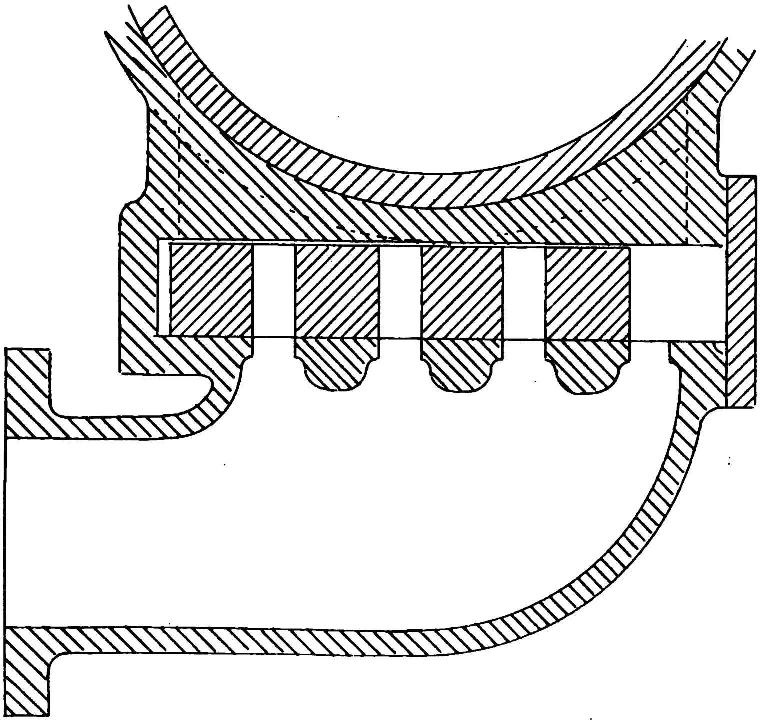
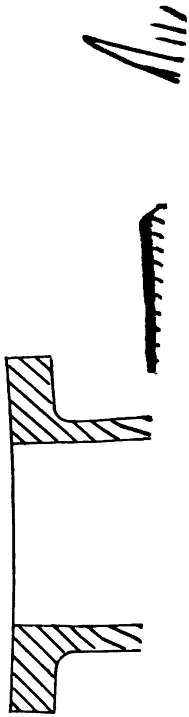


FIG. 145.

engine fitted with "The Richardson" cut-off gear and governor.

Valve Driving Gear. The general arrangement can be best seen from the end elevation and plan, Figs. 148 and 149. In the plan, Fig. 149, the valve gear driving shafts, *A* and *A*, are clearly shown; these revolve in equal times with the engine shaft, and were formerly driven by a pair of equal sized bevel wheels, as in Fig. 150. The one on the crank shaft had to be of large size owing to size of shaft, and the one on valve shaft had to be of corresponding size so as to revolve in the same time. Many manufacturers still use bevel geared wheels for the purpose, but they have a tendency to be noisy. By using machine cut wheels, all jar and rattle

Before co
examine Fi
working d
end elevati



engine fit

Valve
Driving
Gear.

shafts, A
times wi
pair of ec
crank sh
the one
revolve
bevel ge
to be nc

can be prevented, but, owing to the large size of the wheel on the valve shaft, there is always a tendency to ring, a tendency

Skew Gear. which pads of wood and indiarubber cannot altogether counteract. Some years ago Mr. John Buck, one of the author's assistants, suggested to him the

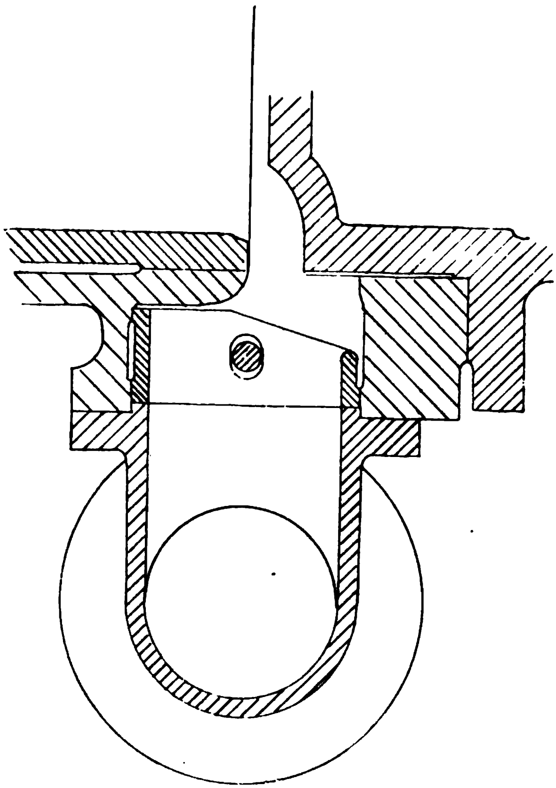


Fig. 146.

use of skew gear, and this has been adopted with very great success. Fig. 150 shows the usual bevel gear, and Fig. 151 the skew gear now used in preference.

There is an apparent mystery about differential skew gear, but in reality it is very simple. The one on the crank shaft has to be of large size owing to the size of shaft, but it would be inconvenient, and it is quite unnecessary, to make the valve shaft wheel of the same size. All that it needs, is to have the same number of teeth as it has, and it will then revolve

at the same speed. Apparently, as seen in Fig. 151, there is a difference in the pitch of the two wheels *A* and *B*, and, as measured on the face shown in the drawing, one is exactly double the pitch of the other, but, measured at right angles to the face of the teeth, they are exactly the same. This is

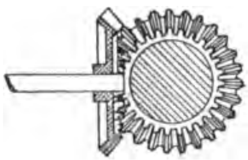


Fig. 150.

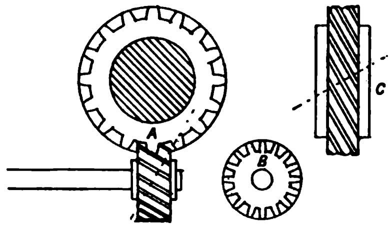


Fig. 151.

more clearly shown by the side near *C*, with the dotted line drawn at right angles to teeth. In the drawing, Fig. 148, the skew gear is shown covered up by a casing, the bottom part of which holds a bath of oil. The gear works very sweetly and easily and is quite silent.[1]

The way in which the valve excentrics are mounted on the valve gear shaft is most clearly shown in the plan, Fig. 149,

[1] If any reader is still puzzled about the skew gear driving from a large wheel to a small one at an equal number of revolutions, the following will make it quite clear.

Proportions of Skew Wheel.

If a single threaded screw gears into a toothed wheel which fits its thread and which has 32 teeth, it is very evident that the screw shaft would have to go round 32 times in order to make the wheel revolve once. It is also equally evident that the diameter of the screw shaft makes no difference, it may be large or small, the only thing which matters is the number of threads on the screwed shaft, thus—

With 1 thread	in shaft	it would	have to	revolve	32	times	to	1	of	wheel.
2 threads	,,	,,	,,	,,	,,	16	,,	,,	1	,,	,,
4	,,	,,	,,	,,	,,	,,	8	,,	,,	1	,,	,,
8	,,	,,	,,	,,	,,	,,	4	,,	,,	1	,,	,,
16	,,	,,	,,	,,	,,	,,	2	,,	,,	1	,,	,,
32	,,	,,	,,	,,	,,	,,	1	,,	,,	1	,,	,,

Now a skew wheel is merely a many threaded screwed shaft of short length, and for some sizes the simplest and cheapest way of making them is to turn a long shaft the diameter required, cut the number of threads required upon it, and then saw it into short lengths, each of which will be a perfect skew wheel. Skew wheels may work at any angle with each other, but where required to work at right angles, all that is necessary is to find the angle of the driver, subtract this from 90, and the remainder will be the angle of the driven one; thus if one be 60°, the other will be 30°.

and it is there seen that advantage is taken of the same shaft from which to drive the governors, which are most conveniently mounted in the centre of the steam chest which forms part of the cylinder.

Separate Excentric for each Valve. By having a separate excentric and rod to each valve, the most perfect adjustment of all the valves can be got much more easily and perfectly than when they are all connected together.

As we have got our valve shaft revolving at the same speed as the crank shaft, we can fix our excentrics just in the same way as if we were fixing them on the crank shaft. We have, however, the advantage that the excentric sheaves and straps may be much smaller, and the rods much shorter than if we drove from the crank shaft. Further, with equilibrium valves the work the excentric has to do is very light comparatively, and the travel is very small. If we give a travel equal to

Drop Valve Opening. $\frac{1}{8}$ of the diameter of the admission valve, this is quite sufficient opening for it; this would give us $\frac{3}{4}$ of an inch for a 6 inch valve, and if the excentric has a travel of one inch, that is all we need.

In considering the action of this valve gear we need not further notice in detail the action of the steam piston, as our previous illustrations have made us familiar with the relative motions of crank and piston, and this is the same whatever the form of valve gear.

What we want with any form of valve is to leave the steam port closed during nearly the whole of the exhaust stroke, lift the valve suddenly at, or slightly before, the exhaust stroke is finished, open the valve quickly and widely for the steam stroke, and hold it wide open for as long a time as possible during the steam stroke, releasing and closing it suddenly when we get within $\frac{7}{8}$ of the end.

In order to be able to do this the centre line of excentric must be nearly in a line with the crank, and travel along with it. The position found in practice to be the best is that shown in Fig. 152. Here *A* is the crank, and *B* the centre of excentric, 30 degrees in advance of the crank centre line. By the time the crank has got into position 1, $\frac{7}{8}$ from end of exhaust stroke and going in the direction of arrow, the valve gear tripper will have reached the end of its stroke and have engaged with the valve lever as shown in Fig. 135. By the time crank

has got into position 2 any slack (of which there is very little) will have been taken up, and the valve will have just opened, as with drop valves there is no lap to travel over before opening, as in any kind of slide valve, including the Corliss..

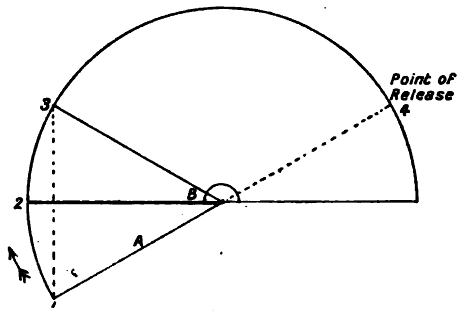


FIG. 152.

Thus with the small advance we have engaged the gear and opened the admission valve at the very commencement of the stroke. By the time the crank has got to position 3 the admission valve is nearly half open, and this opening is fully sufficient to keep up steam practically to steam pipe pressure. The piston by now has only advanced $\frac{1}{16}$ of its stroke. As the crank advances and the piston attains greater speed, the valve continues to open wider, and will be held wide open to nearly the end of the stroke at 4, if not released by the releasing gear earlier.

Full Admission.

The form of diagram obtained from such an admission is almost a complete rectangle, Fig. 153 being a copy of an

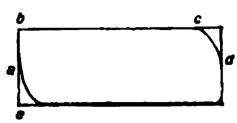


FIG. 153.

actual diagram taken from such an engine, *a* to *b* being the lead, *b* to *c* the admission, *c* the point of cut-off, *c* to *d* the expansion, *d* the point of exhaust, and *d* to *e* the exhaust line, *e* to *a* the compression.

It is only with a drop valve admission that it is possible to get such a diagram combined with a releasing gear. Of course, such a distribution of steam is very wasteful if con-

tinued for many revolutions, but the possibility of such a perfectly full admission is of great value for engines driving the dynamos of an electric tramway system, where immense variations in the load take place, and when such a sudden addition to the power of an engine can be made simultaneous with the application of a sudden but temporary load. By this means, a practically uniform electric current can be maintained; this will, however, be dealt with more fully under the head of governing.

Valve Lead. So far we have dealt with all our valves as if the opening and admission took place exactly at the commencement of the stroke, and what is known as "lead" has only been incidentally referred to. In the case of slow moving pistons to admit exactly at the commencement of the stroke is the best thing to do. When, however, the piston moves rapidly it is necessary to open the admission valve a little before the piston arrives at the end of the cylinder. This is done that we may be sure to get the full boiler pressure at the beginning of the stroke. This is easily accomplished by giving a little more forward motion to the admission valve excentric, or "advancing" it, as it is termed; in other words, simply turning it round on its shaft in the direction of its motion.

Angular Advance. In the case of slide valves with a lap equal to the width of the port and a travel equal to four ports, then without any lead, the angle at which the excentric is set is 120°, as shown in Fig. 154. If, however, we require the $\frac{1}{8}$ of a port lead, then let a line be drawn downwards from *a* and a distance equal to $\frac{1}{8}$ be marked out on centre line, and

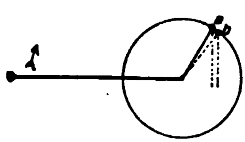


FIG. 154.

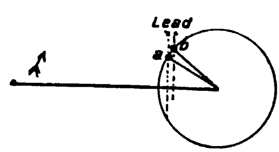


FIG. 155.

another line be projected upwards to the excentric circle, and this will give the point *b*, at which the excentric must be fixed for it to give this amount of lead. The full lines in the diagram show the position of excentric without lead, and the dotted line with $\frac{1}{8}$ of a port lead. It is evident that if the travel of valve remains the same, the port will be wide open $\frac{1}{8}$ of a half

revolution earlier, and that it will close and exhaust earlier in the same degree. All this, however, is necessary for the successful working of high speed engines. There are other and more complicated ways of getting lead to valves and setting out excentrics on the shaft, but none more convenient and simple than that shown.

Fig. 155 shows how to give lead to a drop valve, *a* being the position of excentric centre without lead, and *b* the position with lead, the distance between the dotted lines showing the amount.

In Fig. 156, *a* shows the same position of valve excentric centre as in Fig. 155, but such a position can never be used in connexion with any valve which has lap like a slide or Corliss valve has. From the point *a* any advance of the centre must take place first equal to the amount of lap, i.e. from *a* to *b*, and

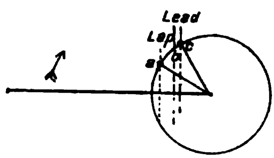


FIG. 156.

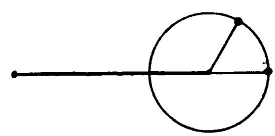


FIG. 157.

then from *b* to *c* for the lead; this will bring the excentric centre to *c*, and then it is seen that a Corliss valve cannot give the same wide range of expansion which can be got with a drop valve, which, having no lap, opens with the first movement of the valve gear in that direction.

Excessive Lead. While suitable lead is useful, $\frac{1}{8}$ to $\frac{1}{4}$ of a port is generally found sufficient, and care must be taken that it is not too much, or great loss of power may result. To take an extreme case, but one which the author has seen in actual operation, where the excentric has been advanced so far that its centre became quite opposite to the engine crank, as in Fig. 157, thus giving a full port lead.

It may be thought at first that an engine would not run at all under such conditions, but it will run and run quite smoothly in either direction in which it is started, but with comparatively little power. The sort of diagram which it gives is shown in Fig. 157; assuming it to work with 100 lb. of steam. It will cut off the admission at $\frac{1}{4}$ stroke at *A*, and expand down to

50 lb. by the time it reaches ½ stroke at *B*. At this time the exhaust port will begin to open, and the bulk of the steam will rush out without doing any work, but, as it cannot all get away at once, the pressure will fall approximately on the line *B C*, *C* being the atmosphere line. The exhaust port remains open up to *D*, when it closes, and the steam in the cylinder will compress, rising in pressure from *D* to *E*, when it meets the entering steam, due to the great lead. The rush of entering steam meeting the piston going in the opposite direction, will drive the pressure up above that of the steam pipe, though not so high as the indicator shows, that being partly due to the action of its spring. Still it does rise above that of the steam pipe from *E* to *F*, and then as the piston comes to rest falls to the normal pressure, making the loop *G*.

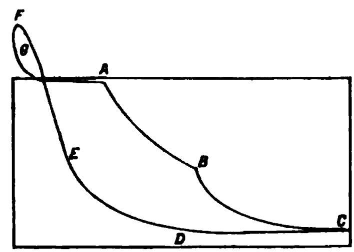


Fig. 158.[1]

It is needless to say that such a set of the valves should never be resorted to, as it is very wasteful both in power and steam.

We shall show shortly how sufficient compression can be got without excessive lead.

Free Exhaust. It is very important to have a free exhaust, but instead of resorting to exhaust lead a better way is to make the exhaust ports and passages sufficiently large after the valve is passed, and to avoid as far as possible all bends. Where these cannot be avoided they must be made as easy and gradual as possible, and somewhat larger than the exhaust pipe itself.

While exhausting at high pressure, like most locomotives do, the above precaution and sometimes even exhaust lead or negative lap, has to be resorted to; yet, on the other hand, when exhausting at very low pressure, i.e. much below the atmosphere, we have an opposite state of things to consider.

Compression. A little compression before the end of the stroke is very useful, and is conducive to easy running. It

[1] This diagram is for illustration only. In actual working, the exhaust and steam lines would be much nearer together. Were there no loss of heat in the cylinder they would be parallel.

avoids that sudden blow which the piston would otherwise receive on admission of steam. That blow means a "knock" in the engine every time it turns the centre, unless something be done to prevent it. However well bearings may fit, and however perfectly they may be adjusted, there is always some space between the sides of the main bearing and the crank shaft, and especially between the bearings at each end of the connecting rod and the crosshead at one end, and the crank pin at the other, and it is surprising what a sharp shock or knock can be caused by the sudden admission of steam bringing these surfaces close together.

Now if instead of this we can have a pressure imperceptible at first, but gradually though rapidly rising to a considerable amount, the knock is avoided, and all the bearings are brought into a position suitable for receiving the steam pressure and making another stroke in the opposite direction.

Of one thing we may be quite certain, viz. that if we have nothing in the cylinder we can compress nothing; with non-condensing engines we always have something, at least the 15 lb. pressure of the atmosphere, and generally a little more. Thus with a steam gauge pressure of 2 lb., i.e. 17 above a vacuum, and closing the exhaust port at $\frac{7}{8}$ of the exhaust stroke, we have $\frac{1}{8}$ of a cylinder full of steam at 17 lb. pressure. By the time the piston has reduced the space by one-half, i.e. $\frac{1}{16}$ from end, the pressure will have doubled to 34 lb.; when again reduced by one-half, i.e. $\frac{1}{32}$, the pressure will have again doubled to 68 lb., and if reduced still to another half the pressure would be 136 lb., and this is what would take place if we had a perfectly fitting piston working in a cylinder with no clearance, i.e. no space between the piston and cylinder cover when at the end of the stroke. There is, however, always some such space, though for the sake of getting suitable compression and also for economizing the steam it should be as small as possible.

Supposing that in the case of a 24-inch cylinder with 48-inch stroke we had reduced the clearance to $\frac{1}{4}$ of an inch at each end, it would be a fairly good clearance, about as small as it is safe to make it; let the compression begin at 3 inches from the end of the stroke, then the real space in which the steam will be compressed will be $3\frac{1}{4}$ inches plus the cubic capacity of the steam and exhaust ports, so far

as they communicate with the cylinder; hence the need for making the space as small as possible. In actual practice these ports have to be taken into account, but this volume varies much with different classes of valve, and for simplicity of illustration we will leave these out of account at present. We have, therefore, 3¼ inches filled with steam at 2 lb. above atmosphere, or 17 lb. absolute. Now, when the piston has gone another 1½ inches the pressure will not be doubled as it would be if there were no clearance, but only increased in the same proportion that the space is diminished. Assuming we start with 3¼ inches and finish with ¼ inch, or $\frac{1}{13}$ of the space, the pressure will then be increased 13 times, and $13 \times 17 = 221$ lb., which again is more than we need and more than we should get, owing to the port clearance. And if the port clearance equal the end clearance, which it well may do, then we should have a space of 3½ inches to compress into ½ an inch, or $\frac{1}{7}$ of the space, and therefore 7 times the pressure, thus $17 \times 7 = 119$ lb., which we can just do with when we have a pressure of 120 lb. to meet, and this steam pressure would come on without any shock and without tending to cause the engine to knock. In fact, if the compression be properly adjusted, the bearings may be comparatively loose, and the author has many times seen such an engine, which at first starting would knock violently owing to loose bearings, but when working at full speed and with full pressure would work and run quite silently.

Compression when Condensing. The above is what we should have with a high pressure non-condensing engine, but if instead of 2 lb. above the atmosphere we have only 2 lb. above a vacuum, i.e. 2 lb. absolute, as is the case when working with a condenser making a good vacuum, then with the same amount of clearance we should have instead of 7 times 17 or 119 lb., only 7 times 2 or 14 lb., and this is not enough for smooth working. The clearance cannot be safely reduced any more; all we can do then is to imprison the steam a little earlier, this is accomplished with slide valve by giving a little inside lap which, if other things remain equal, will also mean a later exhaust unless the excentric be advanced to a corresponding extent, which would again increase the compression, and this is the best thing to do.

When independent exhaust valves are used, whether drop

valves or gridiron, there is no difficulty; they can be set to exhaust at any point and to compress at any point, and this is one of the great advantages of such exhaust valves.

In order to get enough compression with a condensing engine we have to start compressing earlier. Thus if our cylinder had 48 inch stroke, instead of starting to compress at 3 inches from end of stroke, we should start at 9 inches; thus, with a total clearance amounting to ½ inch we should have 19 half inches, and our steam pressure being 2 lb. this multiplied by

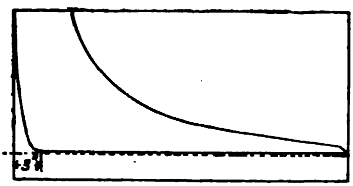


FIG. 159.

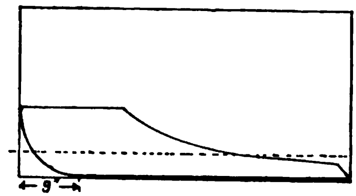


FIG. 160.

19 would give us 38 lb., which is all we need. In the two diagrams, Figs. 159 and 160, the first shows the compression of 17 lb., for 3 inches up to over 100 lb., and the second, Fig. 160, the compression of 2 lb. in a space of 9 inches, up to 38 lb. in a cylinder working with 100 lb. of steam, and representing the low pressure cylinder of a compound engine.

Valve Lubrication. Before leaving the question of exhaust valves, the great advantage of efficient lubrication of slide valves must be pointed out. We saw (p. 116) how great was the pressure upon a slide valve when working with high pressure steam. This can be removed by the equilibrium ring, as we then showed; but this is only in cases of single valves. It is so very difficult to balance slide valves with cut-off plates working on the back that it is seldom attempted, but in the case of compound engines working with, say, 160 lb. pressure, cut-off valves are regularly used, and the slide valve works without any balance ring.

Proper lubrication can, however, very materially reduce the friction. Such high pressure valves are in the best practice made of gun-metal or bronze, or some material which will wear under pressure more rapidly than the cylinder face, as it is much more easy and economical to replace a valve than a cylinder, and when very high pressures were first introduced,

the author has not infrequently seen a cylinder entirely ruined by a few months' work. The difference that a good lubricant properly applied makes to the wear of a valve is strikingly shown in a case to which the author's attention was called and shown in Fig. 161. This is the end view of a cylinder and steam chest in which works the main valve *A* and the cut-off valve *B* in the position as shown; they were lubricated with good oil through a Sight feed lubricator fixed on the steam pipe, the assumption being that (as many contend) the oil on entering with the steam, gets pulverized and mixed so thoroughly with the steam, that no other method of lubrication, either for steam chest or cylinder, is necessary. Everything in this engine went all right with the exception of the main slide valve, which, within three months regular work, got worn to the shape shown in Fig. 161.

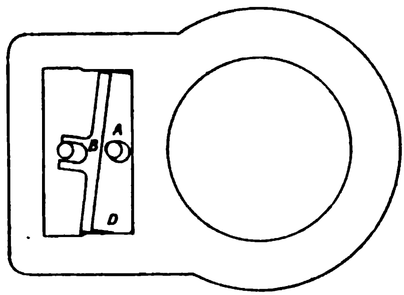


FIG. 161.

Valve Wear.

A new valve was put in, and the same effect took place, when the writer was called in to advise. It was evident on examination that the whole defect was due not to insufficient, but to the defective means of lubrication through the steam pipe, which came in at the top of steam chest at *C*. On comparing the worn valve with a new one, it was found that, measured at *D*, the bottom of the valve, there was no appreciable wear, but that it gradually increased as the bottom was left till it assumed the wedge shape and was spoiled; further, all the wear was on the cylinder face side, as might naturally be expected, owing to the greater pressure due to exhaust cavity in valve. The bottom part of steam chest was also much more oily than higher up, and it was evident that the oil entering at *C* fell nearly all to the bottom and was largely

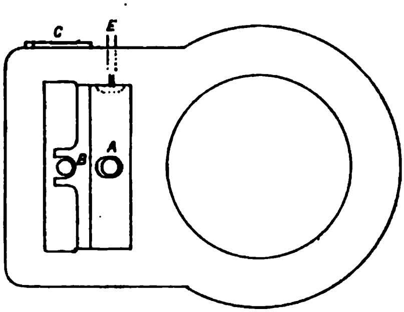


FIG. 162.

wasted. The connexion of the Sight feed lubricator was altered, and the oil was brought in at *E* and dropped into a cavity made in the top of main valve, extending nearly its whole length, as shown in Fig. 163.

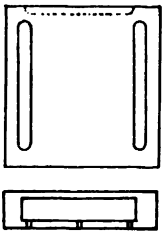


Fig. 163.

From the cavity it escaped through a number of small holes in the side nearest the cylinder, and thus lubricated the valve face thoroughly, passing from the face into the cylinder with the steam.

The alteration made a radical cure, no more valves were worn out in an unreasonable time, much less oil was used, and the internal parts of cylinder were better lubricated.

It is evident from the illustration that if the lubrication of high pressure slide valves be properly done, balancing rings and plates may be dispensed with for all excepting very large quick moving valves.

Piston Valve. There is another class of steam valve we have not yet dealt with known as the piston valve. The elemental principles of piston valves are very simple, and are shown in Fig. 164. Here *A* is a standard slide valve with its two working faces *e e*, and shown in section at *B*. Assuming the valve to be 12 inches deep from *C* to *D*, then a cylinder of 4 inches diameter would be of slightly greater length round it. Now if we could regularly and truly bend this slide valve with its face outwards till the two ends met, it would become a cylinder 4 inches diameter, large at the ends *e e*, and smaller in the middle where the exhaust cavity is. The valve would then have the shape shown in the lower part of the figure at *F F*. The working faces of the cylinder are also curved to correspond, till they become a female cylinder bored to exactly fit the cylindrical valve, as shown at *G G*. In the cylinder the valve works longitudinally. The ports, it will be seen, are curved quite round the piston. If the steam be brought in at the two ends of the valve chest *F F* it will press equally upon

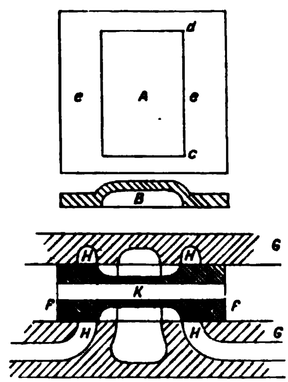


Fig. 164.

the two ends, and, as the steam ports *H H H H* entirely surround the valve, it has equal pressure all round its circumference. Such a valve is in reality in equilibrium with any pressure of steam, and is largely used for the high pressure cylinder of compound and triple expansion engines. The valve shown in Fig. 164 is, as stated above, the simplest elemental form of piston valve, and, with a few shallow grooves turned in its face, may be used, as shown, for comparatively small engines.

Breeches Pipe. The steam with such a valve would be introduced into the steam chest by a breeches pipe cast in the cylinder and steam chest, which are usually made in one piece. Great care has to be exercised in making the core for such a steam chest casting, in order to have the ports perfectly true in distance from each other and size, as it is almost impossible to tool them. Sometimes instead of a breeches pipe being used, the steam is introduced at one end of the steam chest and communicates with the other end through the central passage *K* of the valve. Unless the valve be large enough to give a perfectly free passage of suitable area, it is better to use the breeches pipe, as by its means an equal pressure at each end is secured.

If such a valve be made a really good fit, working as it does with equal pressure all round it, i.e. in equilibrium, if the steam be clean and free from grit, there is practically no wear upon it. When the length of steam chest is limited the collar and nuts of the valve spindle can be let into each end, as shown in Fig. 165.

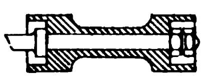


Fig. 165.

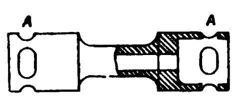


Fig. 166.

Whistling Valve. There is sometimes a curious tendency for such a valve to cause a whistle just as its working edge gets to the edge of the steam port. The whistle continues only for a fraction of a second, but when, as with a quick running engine, it occurs 600 times a minute or so, though it may not be loud, it is far from pleasant. This tendency can be entirely cured by extending the ends of the valve, as shown in Fig. 166, at *A A*, and making a number of slots with curved ends as

shown, through which the steam escapes to the ports. Such a valve works perfectly silently, and, curiously, is much less liable to leak than a valve with solid ends like Figs. 164 and 165. The steam passes from within the valve to the ports in this case, and it seems as if the very slight elasticity of cast-iron yielded under the pressure of the steam just enough to make the surface steam-tight, as it will do if the ends be of the right thinness.

Central Admission. When it is more convenient to do so the steam may be admitted into the middle cavity of the piston valve and exhausted at the ends. A valve and ports drawn to the same scale as Fig. 164 is shown in Fig. 167 which has central admission, the steam entering at *A* and passing in the direction of the arrows to the steam port *B* and exhausting at *C*. When there is a central admission the ports and bars have to be arranged somewhat differently from the end admission, the proportions shown in Fig. 167 give a good working example of how they should be made; the total length of the valve being equal to the distance over the ports from *D* to *D′*. As a matter of course when the steam is admitted at the centre, it has to be exhausted at each end, the two ends being connected either by a breeches pipe outside or by a breeches passage cast in with the steam chest. A central passage in the valve itself if sufficiently large for the free passage of the exhaust steam also answers the same purpose and enables a single exhaust discharge to be used.

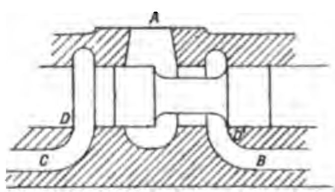


FIG. 167.

Apart from the fact that in some designs of engines it may be more convenient to have a central admission, there is no advantage in it beyond the fact that the valve is a little shorter, and that the pressure at each end is reduced from boiler to exhaust steam pressure, and there is consequently less pressure on gland packing of the valve spindle.

Packed Piston Valve. When piston valves of large size are used it is found necessary to use packing rings in order to keep this sufficiently steam-tight. Such a valve is shown in Fig. 168. They are generally made to work in a sleeve or liners as shown at *A A*. All the parts of the levers

are connected together by bars across the ports, the bars being preferably made diagonal so that there is no risk of catching the rings or wearing in straight grooves.

The liner has a shallow collar *D* at one end, and is firmly held in its seat by means of a ring *B* which just fits between the liner and the cover. The valve is one of the author's designs for a short stroke high revolution engine, and the method shown of holding the lining in position he considers preferable to fixing it by set screws through the collar *D* as is sometimes done.

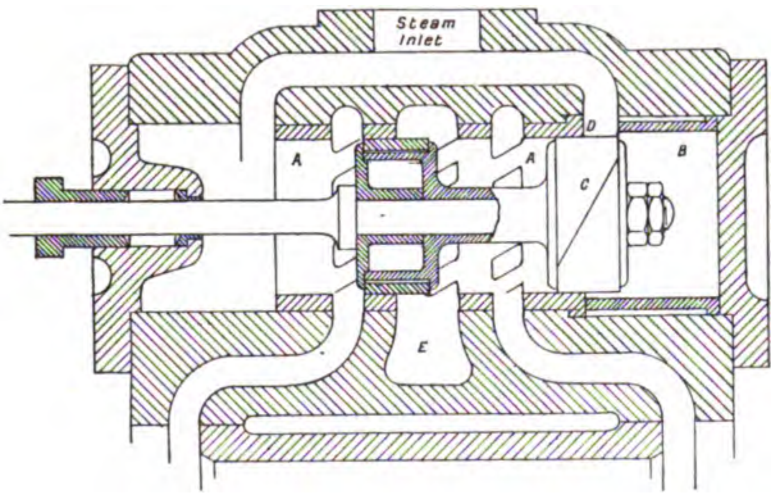


FIG. 168.

When a piston valve is used for a long stroke engine the steam ports may be made as near as convenient to the ends of the cylinder so that they may be as short as possible, and the two working parts of the piston valves separated a corresponding distance from each other by a distance piece between them, the whole of the part between the two pistons being then exhaust cavity.

The exhaust passage *E* leading to exhaust pipe may be taken out at any convenient position between the two pistons. With a slide valve having pressure on its back, we like to keep the exhaust cavity as small as is consistent with free exhaust so as to save pressure on it; but with the piston valve no such precaution is needed, as it is in equilibrium. Also when the

steam ports are wide apart, the linings of the valve cylinder would be in two parts, each inserted from its own end.

As any parts coming loose in a piston valve might cause very serious damage the parts are made as few and simple as possible, and there are no tongues or springs in the piston rings; *C*, shown in section at the other end of valve, is a simple ring of metal bored out, to be when finished a loose fit upon the piston body. A diagonal slot is cut as shown at *C*, the two inclined ends are brought closer together on a suitable mandril and then turned to an exact fit in the valve lining. If proportioned aright such a ring will have enough elasticity to keep it steam-tight in the lining and still be quite easy to move.

Piston Rings. The piston rings are easiest and most economically made by casting of suitable metal a cylinder with a flange on its end, as shown at Fig. 169, boring it out right size, but leaving another cut to take off its surface; then in a screw-cutting lathe, by means of a turning tool, or, better still, a revolving milling cutter, cut the spiral, shown in Fig. 169, nearly through; after which cut off in lengths as shown at 1, 2, 3, 4, and each length cut off will be a perfect valve packing ring. The width of the groove cut needs to be enough to allow for the spring of the valve ring, and it should be about $\frac{1}{25}$ of the diameter of the valve. The ends when put together will fit without any adjustment, and, if held closely together by a clip while the final cut is taken off the surface to make it fit the lining, will when in place prevent any passage of steam.

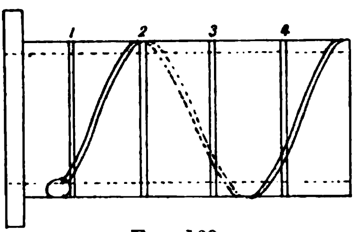


Fig. 169.

There are many other valves, both steam and exhaust, but those illustrated above are typical of all, and if the construction and action of these is well understood by the student, he will have no difficulty with any other.

CHAPTER X

The Governor

THERE are very few steam engines which have always the same amount of work to do every hour of the day, and fewer still which have the same work for each minute of the hour. Even if the load were the same it is unreasonable to expect always the same pressure of steam. The changes in load or pressure may be slight and few in some engines, and great and frequent with others. It is to compensate for these changes that the steam engine governor is required, and its object is to maintain a constant uniform speed under any changes in load or steam pressure.

The apparatus which is used consists generally of a pair of balls hung upon arms, the upper ends of these arms being jointed to a revolving spindle by means of which they are caused to revolve round a common centre.

There are but few students who have not in their boyhood tried to throw stones by means of a sling; here the stone being at the end of the sling has motion given to it, causing it to revolve round the hand of the thrower with greater and greater velocity, till he feels sufficient tension on the strings, when the stone is released, flies off at a tangent to the circle described, with a force proportioned to its weight and velocity.

Centrifugal Force. The power thus generated is what is known as centrifugal force, or the force tending to fly from a centre.

The picturesque but now almost extinct street acrobat used in former days to enlarge his circle of spectators, when they came too close, by holding from his upstretched hand the centre of a long cord from each end of which hung a large soft ball; by a dexterous swing of his hand he set the balls revolving first close to his person, and then, as they gained speed, at a wider angle till they revolved rapidly, nearly horizontally, as

shown in Fig. 170. The force which maintained them in that position so long as they revolved at that speed was the same centrifugal force.

Whether either of these illustrations suggested themselves to James Watt, when his fertile mind invented the governor, we know not; but the apparatus he invented for controlling the speed of engines was based upon the same idea. Having got the idea of revolving balls rising constantly higher with increase of speed, all that was needed was to connect them in some way with a valve, which should close as they rose and open as they fell, and this is what was done.

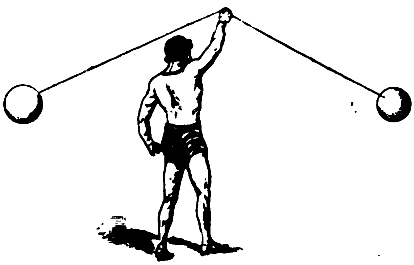


Fig. 170.

Watt's governor is shown in Fig. 171. To each of the revolving arms a jointed link is attached, also joined at the lower end to the sleeve *A*; this sleeve is free to slide up and down as far as the links will allow it; it is evident, therefore, that as the governor balls rise, the sleeve will rise also. A forked lever fitting between the collars of the sleeve, rises with it, and, by means of suitable cranks and rods, opens and shuts the butterfly valve shown in Fig. 172.

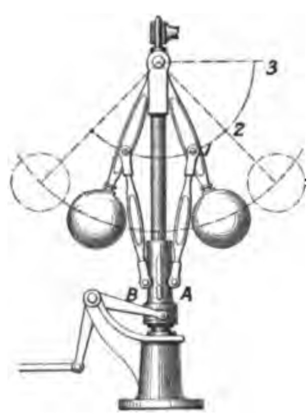


Fig. 171.

But little examination of Fig. 171 will be needed to show that whatever speed of revolution is necessary to cause the balls to move from position 1, a greater speed is necessary to cause them to rise to position 2; a much greater speed still would be required to move them much above that position, while an infinite speed could not raise them above position 3.

The Pendulum. Most people are familiar with the oscillation of the pendulum of a clock, and many will have noted

the fact that the shorter the pendulum is, the more frequent is the number of the oscillations—that is, the faster the clock ticks.

The laws which regulate the oscillation of a pendulum also rule when the suspended weight revolves, instead, in a circle. Thus a pendulum 39·139 inches long, will oscillate from right to left in one second, thus making 60 oscillations or vibrations in a minute. This is called a seconds pendulum, and, though owing to our antiquated measures it is a rather awkward number to use, yet we have to take this number, 39·139,[1] as our unit in all our calculations about pendulums. If, instead of swinging the pendulum backwards and forwards, we swing it round in a circle, when the point of suspension is 39·139 inches above the plane of rotation, it will make 30 complete revolutions, equivalent to 60 oscillations in one minute; this is independent of the weight of the ball, or the size of the circle it goes round; all depends upon the height of the point of suspension from the plane of rotation. Therefore the number of revolutions varies with the height, but not directly so. The law is as follows:—"The number of revolutions (a double oscillation) is inversely proportioned to the square root of the height of the cone of revolution."

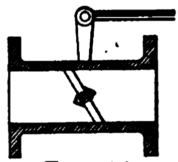


FIG. 172.

Law of Oscillations.

Now the square roots of our standard seconds pendulum (39·139 inches) is 6·2561. If then we want to find the time of revolution of another length, say 25 inches, we proceed as follows: the square root of 25 is 5; the number of revolutions the pendulum would make would be $\frac{6\cdot2561}{5}$ or 1·2512 more revolutions than 30, which was made with the large pendulum, i.e. $1\cdot2512 \times 30 = 37\cdot566$ revolutions per minute. In the same proportion the *time* of each vibration will be smaller, i.e. as the square root of 25 is smaller than the square root of 39·139, so will the time of each vibration be smaller, i.e. as 5 is to 6·256; now $\frac{5}{6\cdot256} = \cdot799$ of a second for each vibration or semi-revolution and, of course, double this amount, or 1·598 for a complete revolution. Then $60 \div 1\cdot598 = 37\cdot53$ revolutions a minute instead of 30 with the larger pendulum.

[1] The exact number is 39·1393.

Recapitulating our rules we learn that to find the *time* of vibration of a pendulum of a given length we must divide the square root of its length by the square root of the standard pendulum, 39·139 inches long, which is 6·2561, and the quotient will be the *time* of a single vibration in seconds; and, conversely, to find the *number* of vibrations per second of a pendulum of given length in inches. Divide the square root of 39·139, i.e. 6·2561, by the square root of the length of the given pendulum, and the quotient will be the number of vibrations per second of the given pendulum.

Now so far as *vibrations* are concerned the distance they go from right to left and then the height to which the body or weight on the end of pendulum rises makes (within wide limits) no difference in the number of *vibrations* in a given time; this is because at each vibration the weight comes down to the same vertical distance from the point of suspension. Yet, when the weight is *revolving* round a central point with a pendulum of given length, then the greater the height at which the weight revolves the greater is the speed required to keep the weight revolving at that height. This can be easily tested by counting the vibrations of a weight suspended from a pendulum or cord of a given length, and it will be seen that whether the vibrations are to the extent of 1 inch or 20 inches the number of them in a given time is the same. Then let the string be taken in the hand and the weight swing round in a horizontal plane, and we shall find how much more rapidly it has to be swung round in order to maintain it always at the extreme height to which it formerly went at the end of each vibration. This brings us back to our first rule, viz. the number of revolutions is inversely proportioned to the square root of the *height* of the cone of revolution.

Rule for Revolution.

Having mastered these elementary facts, we can see that while the Watt governor was an exceedingly clever invention, sufficient to prevent our engine "running away" when all the load was removed and to regulate its speed fairly well with small changes of load, yet when a wide range of motion is required in a governor, this one requires too much change in speed to supply it. For instance, a pair of standard seconds pendulum with weights at their ends, as shown in Fig. 173, revolves as we have shown at the rate of 30 revolutions per minute.

When the arms are extended till the plane of rotation *a* is round to 25 inches from the point of suspension or the apex of the cone of revolution *P*, as in Fig. 174, they require 37·5

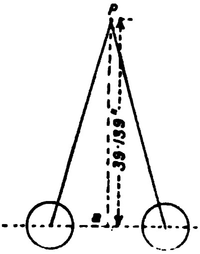


Fig. 173.

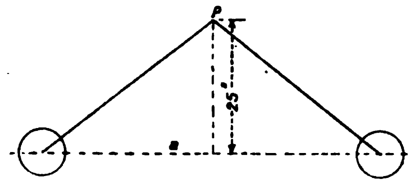


Fig. 174.

revolutions to maintain them in that position, an increase of speed of no less than 28 per cent.; while, when raised to the position shown in Fig. 175, when the plane of rotation is 16 inches below the point of suspension, a speed of 46 revolutions per minute is required—an increase of over 50 per cent.

Butterfly Valve. No such variation in position is required in a governor controlling an engine through a butterfly throttle valve; as such a valve works always nearly closed and this wire draws the steam, and a large movement of the governor is not needed if it be made big enough. It was therefore suitable for low pressures and moderate variations in load. The modern steam engine, however, has to deal with very high pressures and great variations of load, and needs a governor which is quick and powerful and can go through its extreme range with a variation of speed not exceeding 2 per cent.

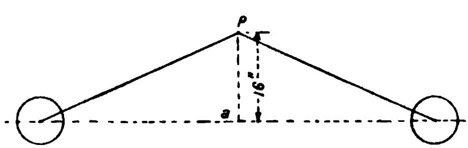


Fig. 175.

Porter Governor. The Porter governor (Fig. 176) is a marked improvement upon the Watt governor. A heavy central weight *A* offers a constant resistance to the lift of the balls which are thus caused to revolve at a much higher speed than would be required without the central weight. The variation

in speed due to the alteration of height of the plane of revolution remains, but if, because of the weight the speed is doubled, the percentage of variation is reduced by ½, other things remaining equal. This governor gives a much wider range of

FIG. 176.

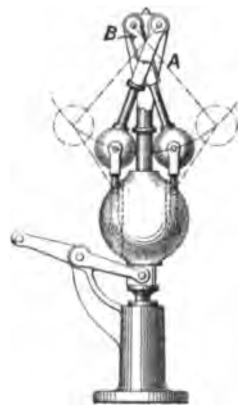


FIG. 177.

travel with a smaller difference of speed than the Watt governor, and has been very largely used both in connexion with throttle valves and also with some forms of expansion gear, notably the Porter Allan.[1]

Cross Arms Governor.

A form of governor possessing some further advantages was designed by the author about the year 1875, and is shown in Fig. 177. In this governor the arms instead of being suspended from a single fixed point, are suspended from two points and cross each other. (Soon after Watt's invention this method had been adopted by various engineers.) The effective length of the arms or pendulums, being that between the centre of the balls and the point *A* where the arms intersect, increases as the balls rise, the increase in length being between the points *A* and *B*. The special feature in this governor differentiating it from the Porter was the close position of the balls at the lowest position; this, combined with the crossed arm, caused the effective

[1] In the following chapter dealing with governors there is incorporated, by kind permission of the Council, the greater part of the paper on "The Mechanical and Electrical Control of Steam Engines," read by the author at the Institute of Civil Engineers in 1895, and awarded the Crampton Prize for that year (*Proceedings of Civil Engineers*, vol. cxx).

length of arm to increase more rapidly than the rise of the balls.

It is not difficult to proportion governors of this class so that the sleeve will have a large range of motion with very small difference in speed. While this is a great advantage it is obtained only at a sacrifice in power, and when rendered isochronous,[1] such a governor is practically useless for controlling any valve or gear which requires much power to move it, as at a fixed speed the weighted ball will stand indifferently at any point between its lowest and highest positions. To be of the greatest use, a governor must have great lifting and forcing power, i.e. power to push down as well as lift up. It must have stability at every point in its range, so that with a certain speed it will stand at that point and no other unless great force is used to move it away, and it must have a large motion of the sleeve in proportion to the difference in speed of revolution.

Stability. The ordinary pendulum governor is stable, but unless it is of great size in relation to the engine, it has but little lifting and forcing power; and in the paper communicated to the Institution by Professor Dwelshauvers-Dery,[2] he states that as a result of experiments, "The logical conclusion is that the best governor cannot, when the greatest fluctuations in the load of the engine occur, prevent the velocity from varying 'by at least 10 per cent.'" These conclusions apply to an ordinary pendulum governor, and are no doubt correct, but much better results can be obtained by the combination of such a governor with a spring.

Spring Governors. In such a combination an entirely new set of conditions is presented. Gravity, which is the only force tending to cause the balls to close or fall in the Watt governor, may now be almost disregarded, for a much stronger force is substituted for it, i.e. for a weight of 50 lb. may be substituted a spring pressing with a force of 500 lb. Such a governor can therefore act equally well in any position. One of the earliest examples of this kind of governor is that designed by the author in 1869, shown in Fig. 178. In addition to the spring another feature was introduced, viz. the length of the arms

[1] i.e. equal in time of revolution at all points; from *isos*, equal, and *chronos*, time.

[2] Minute of *Proceedings Institute Civil Engineers*, vol. cx, p. 277.

was so proportioned that the balls diverged in nearly a straight line for a considerable distance. The form in which the governor was originally used was horizontal, as shown in Fig. 179,[1] and gravity plays so small a part in the working of the governor, that it can be used indifferently in any position. When used in this position the fixed sleeve of the governor had a couple of hollow arms *c c* cast with it to contain the balls, and take all racking strains from the joints of the arms; this is only needed when the balls are of considerable weight and the revolutions few.

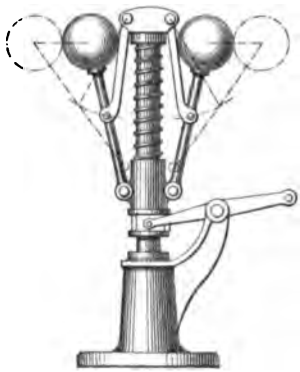


FIG. 178.

We will assume a governor making 200 revolutions per minute and with balls of 20 lbs. weight each (about 5½ diameter) standing when at rest with its balls at a radius or distance from the centre, of 6 inches and flying out to 10·8 inches, the speed being 200 to 204 revolutions per minute, a variation of 2 per cent., we have to find the centrifugal force generated at these two speeds and positions in order to get the strength of the spring to balance it.

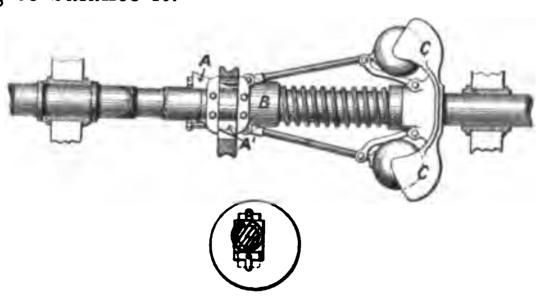


FIG. 179.

The formula for such a calculation is as follows :—

Formula for Revolution.

Let W=the weight in lb. of the revolving body.

,, N=the number of revolutions per minute.

,, R=the radius in feet from centre of motion to the centre of gravity of the revolving body.

[1] Patented Dec. 1, 1869.

Then F the centrifugal force will $= \cdot 00033\ W\ R\ N^2$. From this it will be seen that the centrifugal force varies directly as the weight, or the radius, but in accordance with the square of the number of revolutions. In our illustration—

$W = 40$ lb., i.e. the weight of the two balls.

$N = 40{,}000$ the square of 200 for one position and 41,616 the square of 204 for the other position.

$R = \cdot 5$ of a foot for the closed position and $\cdot 9$ of a foot for the opened position.

We shall then have the calculation—

$$\cdot 00033 \times 40 \times \cdot 5 \times 40{,}000 = 266 \text{ lb. and}$$
$$\cdot 00033 \times 40 \times \cdot 9 \times 41{,}416 = 494 \text{ lb.}$$

The former being the centrifugal force when the balls are closed, and the latter the force when they are fully open.

Assuming that the governor arms are so arranged that we have a travel of 3 inches on the governor slide and that the distance between the fixed and moving sleeves when the balls are closed is 10 inches, we shall need a spring which, when compressed to 10 inches, will have a force of 266 lb., and when further compressed to 7 inches will exert a force of 494 lb., a difference of 228 lb.

The total weight of such a governor need not be more than 80 lb., but it has a lifting and forcing power of 228 lb. between its extreme positions. Such a governor is also very stable, for while moving the whole 3 inches with a difference of speed of only 2 per cent., it would require not less than 9·5 lb. to move it $\frac{1}{8}$ of an inch in either direction from any position it naturally took up, and, of course, double this amount, viz. 19 lb. to move it $\frac{1}{4}$ of an inch. It nevertheless responds immediately to the slightest changes in speed.

A variety of this type of governor, shown in Figs. 177 and 178, where the balls are arranged to expand in a nearly straight line (though doubtless an independent invention) was introduced by Dr. Proell about the year 1884, and is illustrated in Fig. 180. In this governor as originally constructed a heavy weight is used instead of a spring, though the action of the arms is similar. This weight is lifted at an accelerated speed as the balls diverge. Such a governor is extremely sensitive, but has practically no lifting or forcing power, and needs a dash-pot to give it stability. Something in the nature of a dash-pot is introduced in the

Sensibility.

governor and is shown in the section (Fig. 181), but as this contains air only, it is not very effective. A perfect governor does not need a dash-pot, but if one be used it should contain some incompressible substance which can escape only through

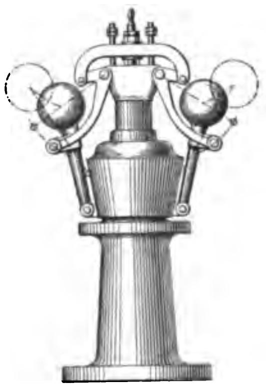


Fig. 180.

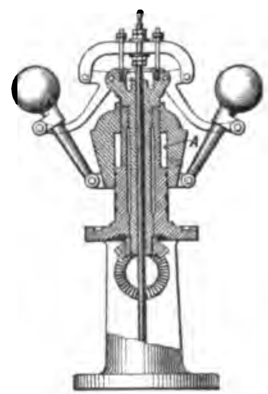


Fig. 181.

a small aperture from one side of a piston to the other, as shown in Fig. 182; here a pipe communicates from the two ends of the cylinder in the middle of which is the cock *A*, when fully open, the contained oil can pass freely, and the cock can be regulated so as to give just the stability required. In the opinion of the author, however, all such attempts to give stability to a non-stable governor are but expedients to minimize an evil which should be, and can be, entirely eliminated.

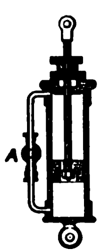


Fig. 182.

Hunting.

A very common difficulty with incorrectly designed governors is that of hunting, i.e. the engine will attain its proper speed of say 150 revolutions per minute with the balls entirely closed, a reduction of load takes place, the engine starts running faster till it gets to say 160 revolutions, when the balls, which have begun to rise suddenly, fly to the top position, tending to shut off steam; this is done and the speed slackens, but the balls do not fall till the engine has slowed down below 150 when they quickly drop and this amusing but very inconvenient process is repeated at frequent intervals. Dash-pots are often introduced in

order to remedy this evil, and they do make the fluctuations spread over a much longer time and thus occur less frequently; more often the governor is thrown out of use altogether, and the regulation done by hand.

This hunting would occur with the Richardson governor discussed above, Fig. 179, if the springs were not properly proportioned, thus if instead of the spring exerting 266 lb. in its extended position, i.e. with balls closed, it exerted 320 lb. then the governors would have to make 220 revolutions before the force of the spring was balanced; any more, however slight, would cause the balls to fly out and they would meet with a diminishing instead of an increasing resistance; so they would expand suddenly and remain in the expanded position until the revolution fell to 202. The spring balancing the centrifugal force at that speed, the least fall below that would rapidly close them owing to the increasing force of the spring, and so they would fluctuate between 220 and 202 revolutions. The successful remedy is to alter the spring by reducing its force to 266 when the balls are closed; any system of dash-pot could only modify the evil but never eliminate it.

Until the last few years governors for controlling the speed were almost universally applied to engines by means of the wing or butterfly throttle valve shown in Fig. 172. This valve, though simple in construction and easy to move, is difficult to fit so as to remain steam-tight. One of the first requisites of a regulating valve is, that it should be able to entirely stop the admission of steam when all the load is off the engine, and this becomes more important as steam pressures become higher.

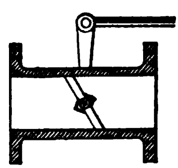


FIG. 172.

The valve shown in Fig. 183 is a considerable improvement upon the butterfly valve. It is easier to make and easier to work, and for merely regulating variations in load with a good governor it acts fairly well; it is in reality an equilibrium slide valve, one cylinder sliding within the other. It is difficult, however, to make such a fit of it as shall exclude steam on the one hand, and leave it sufficiently free to move on the other. Careful experiments with free-running engines show at least 15 per cent. variation between all load on, and all load off, the small space

Slide Throttle Valve.

required to pass sufficient steam in order to drive an unloaded engine being almost incredible.

The Double-Seated Valve.

The double-seated valve shown in Fig. 184 has been found by the author to give the best results of any kind of throttle valve. The arrangement shown is that of a valve not quite in equilibrium when closed, the area of the upper part *A* being greater than the lower part *B* by an amount equal to the area of the valve stalk so that the steam entering in the direction of the arrow presses a little more upwards than downwards when the valve is closed, and does the same when the valve is open, the stalk thus subtracting from the upper area; by this means the valve has a constant upward pressure whether on or off its seat. This makes it more ready to respond to minute

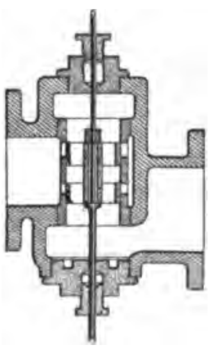


FIG. 183.

FIG. 184.

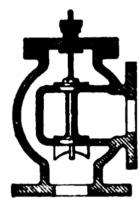


FIG. 185.

changes in the position of the governors, there being no slack or back-lash. Combined with a well-proportioned high-speed spring governor it will, with changes of load not exceeding 75 per cent. from the maximum, maintain a speed within 5 per cent. of the normal; but the difference between maximum load and no load is 10 per cent. with a steam pressure of 100 lb. per square inch, the difference being less with lower pressure of steam.

Another form of valve which is largely used in England and America is shown in Fig. 185. This valve works nearly in equilibrium, the stalks being small and the two discs being of the same diameter. Combined with its own governor of the spring-arm type it will maintain a nearly constant speed with moderate changes of load, but it fails to cut off the

whole of the steam, and thus with light loads or no loads the variation in speed is very considerable. Careful experiments made with it show permanent variation of from 9 to 15 per cent., and temporary fluctuation of from 13 to 25 per cent. these are shown in the table below.

Pressure at Stop-Valve.	Revolutions per Minute.		Permanent Variations in Speed due to Change in Load.	Temporary Fluctuation in Speed due to Change in Load.		Brake H.P.	Time Coming to Rest.
	Light Engine.	Loaded Engine.					
Lbs. per sq. in.			Per cent.	Revs. per Minute.	Dif. per cent.		Secs.
50	345	305	11·5	—	—	5·2	5
60	350	310	11·5	310–360	13·9	6·9	10
70	355	320	9·8	310–370	16·7	8·0	25
80	360	325	9·7	310–370	16·7	9·0	30
90	362	305	15·2	280–370	25·0	10·8	10
100	362	305	15·2	280–370	25·0	11·3	15

All throttle valves, though simple and easy to manage, have necessarily the objection that they regulate the speed by altering the pressure of steam in the steam chest and in the cylinder, and further, that in order to get any good regulation they have to work always with the valve so nearly closed that the full pressure of steam is never available at the full speed of the engine, and thus with a given pressure of steam, less power can be got out of the engine than if there were no throttle valve. Now amongst the essential features of a good governor, one is, that it should not diminish the power of an engine, and another, that it should increase rather than diminish the economy in the use of steam. For many purposes, however, when great delicacy in governing is not needed and when economy is not a primary object, such a valve as is shown in Fig. 184 or 185 is all that is required.

Governing by Admission Valve. The most important type of governor is one which controls the admission valve, and which regulates the speed either by altering the travel and angle of the main valve, or by limiting the admission of steam through controlling the angular advance and travel of a cut-off slide working on the back of the main valve.

Shaft Governors. Governors of this former type generally revolve with the crank-shaft, and are known as shaft-

governors. One of the first of these to be used was the Roby-Richardson shaft governor, first made in 1869 and exhibited at the Smithfield Show in that year. This is shown in Fig. 179. In this case a governor of the spring type is used in connexion with an adjustable excentric shown in Fig. 84. The object to be attained is to move the centre of the excentric in a straight line at right angles to the crank, thus maintaining a constant lead while varying the travel and the point of cut-off. This is accomplished by a pair of wedges, *A A*, connected firmly to the moving slide *B* of the governor and therefore pulled in and pushed out as the slide moves. When the governor balls are closed the wedges are pushed in, and the excentric is then having its full travel, giving a $\frac{3}{4}$ admission of the steam. When the balls expand the wedges are drawn out, and the excentric is moved at right angles to the crank until it has a travel only equal to the lap each way, and therefore admits no steam. The excentric works upon a square made on the crank shaft, which square drives it and the valve, so that there is no strain on the governors, and, while the governor

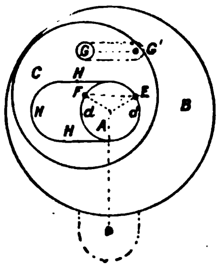


FIG. 84.

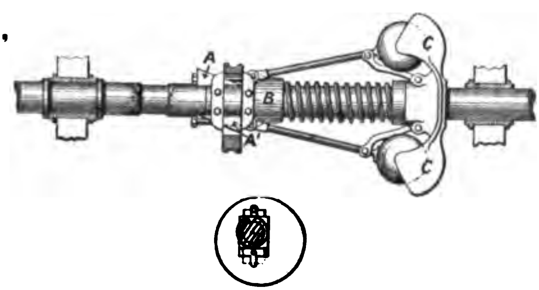


FIG. 179.

with its wedges adjusts its position quite easily with regard to cut-off, they at the same time hold it quite firmly at all points. This is an important point, and permits its application to an ordinary flat slide valve.

The diagram, Fig. 186, is of interest, being an exact copy of the first diagram, so far as the author knows, ever taken from a portable steam engine fitted with automatic cut-off

expansion gear. While well adapted for use with a portable engine, the great length of shaft it required to contain it, prevented its use in many other types of engines.

Fig. 186.

A modification of the governor expansion gear is shown in Fig. 187, and it occupies much less space on the shaft.

In this governor *A* is a disc keyed on to the crank shaft, the position of the crank pin being shown at *C*. In this plate there are planed two shallow recesses in which slide the two plates *B B*. The revolving weights *D D* are carried upon the pins, *E E*, and at the other ends are connected through a sliding joint with the sliding plates *B B*. In each of these plates is cut a diagonal slot, forming that portion of the plate into a wedge. Two projections, one on each side of the excentric, fit into these wedge slots, so that when the weights fly out, the excentric is moved sideways in the same manner as shown with Fig. 179. The springs which counteract the centrifugal force of the balls are shown at *C C*, and act upon the short arm of the weights. The governor occupies much less space, and can be used in positions where Fig. 179 could not.

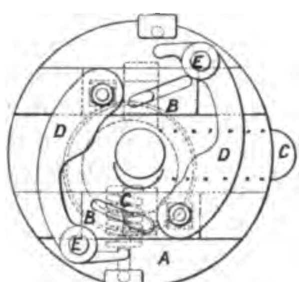


Fig. 187.

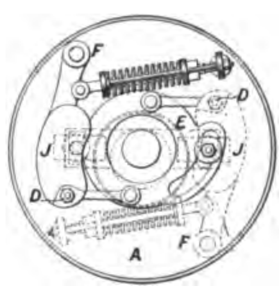


Fig. 188.

A much simplified modification of the arrangement, and one more easily manufactured, has been made by Mr. John Buck,[1] a member of the author's staff, and is illustrated in

[1] Richardson and Buck's patent, No. 1892.

Fig. 188. A plate *A* is keyed on to the crank shaft and it has two slots in a line with each other and at right angles with the line of crank. The excentric has two lugs cut upon it which are tooled to fit into the slots, and the excentric is driven by these lugs and allowed to slide at right angles to the crank, thus varying its travel and point of cut-off and keeping its lead constant. The excentric remains on one side of its driving plate, and on the other side is a turned boss upon which fits a plate *E*; this plate *E* has at one part a wedge-shaped slot made also in a curve; within the slot a sliding block attached to the projecting lug of the excentric engages, and it will be seen that as this plate *E* makes a partial revolution round the boss of the driving plate, it must cause the excentric to move along its guiding slot, and thus vary the admission of steam. The weights, in this case *J J*, are connected to the driving plate by the fixed pins *F F*, and connected with the wedge plate *E* by the links *D D*. These links pull in opposite directions on opposite sides of the boss, and so, easily turn the plate backwards and forwards as the weights open or close. The compensating springs are easily seen. There is but little space required for the governors, as the whole of it can be contained within the rim of the fly-wheel or driving pulley of the engine.

It should be noticed also that the whole of the driving of the slide-valve is done by the driving disc, and that the excentric is held firmly in position at all times by the slots and wedges, which are at right angles with each other, and that there are nevertheless four points in each revolution when it is quite free to be moved and adjusted by the governors. With a properly proportioned governor of this type there is no difficulty in maintaining a practically constant speed of the engine with ordinary changes of load, and with so wide a range as between $\frac{3}{4}$ and zero, in the admission of the steam. Between maximum load and no load the variation of speed can be reduced to 3 per cent., which is sufficiently near for most purposes, and is a much better result than can be obtained with any kind of throttle valve governing. With this method of governing there is, of course, no wire-drawing of the steam, and thus no reduction in pressure and in the power of the engine; and further, the full boiler pressure being always available the steam can be cut off earlier and the maximum economy gained by its expansion.

The subjoined table shows the result of experiments made with the same engine and under the same conditions, fitted at one time with the shaft governor and at the other time with a throttle valve like Fig. 185 and a spring arm governor, the engine being loaded in the latter case till the speed fell 12·66 per cent. below the speed when unloaded and with the wedge governor, to only 4 per cent. below that speed with the same load.

Governor.	Fig. 185.	Fig. 184.	Fig. 187.
	Revolutions per minute.	Revolutions per minute.	Revolutions per minute.
No load.	338	320	312
Full „	300	300	300
Variation per cent. .	12·66	6·66	4
	29½ B.H.P.		35 B.H.P.

With the more delicate governor and the throttle valve the temporary fluctuations, though less in quantity, are naturally greater when measured in time; in one case amounting to 75 seconds; the extreme variation, however, was only 8·7 per cent. and gradually fell to zero. With the shaft governor, the temporary variation was for a moment 16·8 per cent., but the governor arrived at rest within 25 seconds. Another important difference was found, viz. that with the shaft governor the variation is nearly all upward; thus, if the normal speed is 352 revolutions per minute, when the load is suddenly applied the decrease of speed is only two revolutions per minute, and this variation causes an instantaneous upward rise, gradually coming to rest with decreasing oscillations. On the other hand, with the throttle valve the sudden change is downwards, gradually rising with few oscillations from 310 to 350.

It should be noted that all the changes were made suddenly, all the loads being put on or taken off at one time. With gradual changes there is practically no fluctuation, either with the shaft or throttle valve governor.

The following table shows not only the uniformity of speed of an engine with such a governor, but also the great change in the electrical load.

SHOWING UNIFORMITY OF SPEED OF AN ENGINE FITTED WITH A SHAFT-GOVERNOR.

	Revolutions per Minute.	Ampères.	Revolutions per Minute.	Ampères.	
Increasing Load	214	0	208	380	Decreasing Load
	213	50	209	280	
	213	65	211	180	
	210	80	213	80	
	209	180	214	0	
	209	280			

From this it will be seen that the total variation in speed is slightly under 3 per cent. from no load to full load, and that with variations of over 50 per cent., i.e. from 280 to 380 ampères, the variation in speed is only one revolution per minute, or less than ½ per cent. The power of the governor to prevent any of its parts being moved out of their natural positions, is very considerable, the springs requiring a force of 1,100 lb. to move them from the extreme inward to extreme outward position.

It may be pointed out here that the position of the weights and the strength of the springs does not work out strictly according to the recognized laws of centrifugal force. These laws are of course correct for freely moving bodies, but are not true for bodies moving freely at one part and suspended at another, from a point fixed with relation to the centre round which they revolve. The diagram (Fig. 189) shows graphically the relation between the calculated, experimental and theoretical pressures of a spring resisting a certain centrifugal force generated by a weight revolving at a constant speed but at different radii. The slightly curved short-dotted lines show the variation in pressure which experiment indicates is actually required when the weights are suspended on pins, and the straight line shows the calculated strength of the spring when allowance is made for this. The author is indebted to his assistant, Mr. Wansbrough, for assistance in working out this method of calculation, and the diagram shows that in a large number of cases it is correct. It will be noted that the lines intersect before the point of highest pressure, and the

extra strength of spring beyond this point exactly corresponds with the 3 per cent. increase of speed allowed with no load.

Link Expansion Gear. Link expansion gear is chiefly identified with the name of Allan, and was first introduced into Europe at the French Exhibition of 1864. Allan worked, as is well known, with one excentric, in the strap of which was a curved slot. He got very good results, and may fairly be looked upon as the pioneer in slide valve cut-off gear.

There are many varieties of governors and valves now used for the purpose, amongst which that of Hartnell and Guthrie takes high place, and, in accordance with the law of the sur-

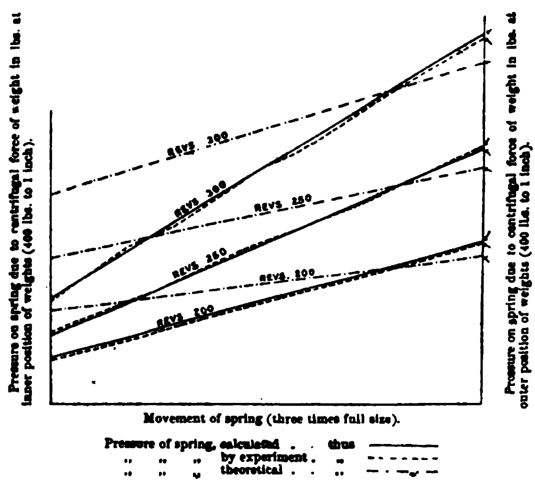


Fig. 189.

vival of the fittest, that which now remains in most regular use, is a combination in which the main valve gives the admission, compression and exhaust, and a supplementary valve on the back worked by a separate excentric and rod, controls the cut-off.

The cut-off excentric rod is divided into two parts, one of which bolted to the strap works a rocking link containing a curved slot, in which a slide block connected to a radius rod works; the other end of radius-rod being coupled direct to the cut-off valve. A link connecting the radius rod to a prolongation of the governor lever causes the radius rod to rise or fall with the governor slide. The excentrics are so

set that as the radius rod rises, the travel of cut-off valve is reduced, and the steam ports are covered by the cut-off valve earlier; and, as the radius rod falls to the bottom of the link, the ports are not covered at all. A well-arranged gear

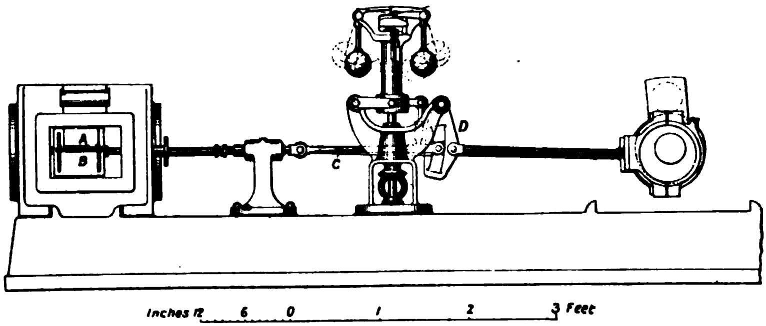


FIG. 190.

of this kind will have a range of cut-off from zero to a little over $\frac{5}{8}$ of the stroke.

A general arrangement of the form of expansion gear is shown in Fig. 190. The steam chest cover is off and shows the half of the cut-off valve *A* on the back of the main valve with its two ports *B*. *C* is the radius rod sliding up and down in the curved link *D*.

Position of Excentric.

Surprise has often been expressed by students that the cut-off in the gear occurs with the shortest travel of the cut-off valve instead of the longest, but a little study of its main lines will show that this should

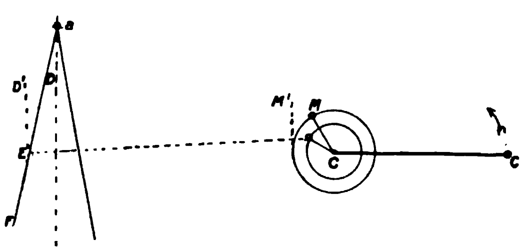


FIG. 191.

be so. While in setting out a working drawing of such an expansion gear many things, such as lap, lead, travel, width of ports and other matters special to the particular engine have to be taken into account, yet the elementary principles

of it will be easily understood by reference to diagram, Fig. 191. The crank *C* is shown just ready to commence a stroke in the direction of the arrow, the main valve excentric centre *M* has to travel a distance *M* to *M′* in order to fully open the steam port. The centre of the cut-off excentric has a smaller travel and greater advance than the main valve. The cut-off link is suspended from the fixed centre *a*, and its centre is in the position *D*, *E*, *F*. Assuming that the length of the cut-off valve equals the distance between the steam ports of the main valve, then when both travel together the action is the same as if there were no back valve, and in the position shown it is nearly in that place, but we shall see that by sliding the radius block from *E* to *D* the back valve will have been moved so far towards the crank that it will have covered the steam port in the main valve, the distance *D′* to *D* being equal to the distance *M* to *M′* the width of the port; in such a position of the radius rod the gear will cut off at zero.

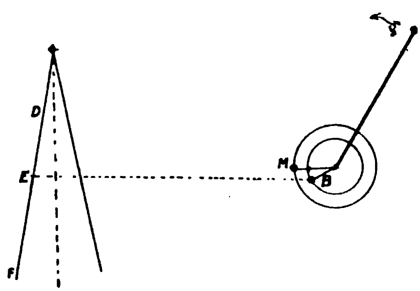


Fig. 192.

When the link is lowered from *E* to *F* the effect is the reverse; this is better seen by reference to diagram (Fig. 192), where the crank *C* is shown in such a position that the main valve is wide open, the centre line of link is still in the same position, the excentric crank being moved the same distance below, that it was formerly above, the centre line of engine. The main valve has now been pushed under the back valve till its port is nearly closed, as the back valve has been practically stationary while the main valve has moved rapidly; but if now we have the radius block from *E* to *F* we shall open the port in main valve and enable steam to be admitted until ⅝ of the stroke is passed. From this time the centre *B* of the cut-off valve is moving at its most rapid rate, and though

the main valve is coming in the same direction, it does not travel nearly so fast at this point and thus we get a rapid cut-off. The diagrams (Figs. 193 and 194) are taken from a compound engine fitted with this gear, and so far as cut-off is concerned, leave little or nothing to be desired.

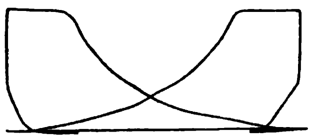


FIG. 193.

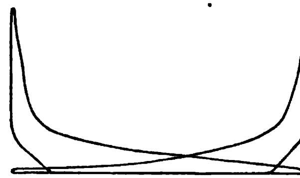


FIG. 194.

Work of Governor. It will be noticed here that the governor has much more to do than merely to close a throttle valve. The pressure of steam on a back valve is not nearly so much as it is on the main valve, but still it is far from nothing, also there is the friction of a fair-sized gland and that of the slide block in the link; all these together make up quite a respectable amount of friction to overcome, and the success of such a gear depends very largely upon the class of governor which is used to work the radius rod. This must have a wide range, great stability, as well as great sensibility, in order not only to cut off properly, but to maintain constant speed.

Dash-Pot. The author has experimented with Watt, Porter and Buss governors in connexion with the gear, but with unsatisfactory results, as all needed a dash-pot to give them stability and to prevent the angle of the link causing oscillation in the governor. The dash-pot arrangement cured this, but of course rendered the governor more sluggish in its action, so that with wide changes of the load there were great temporary changes of speed; though the engine would settle down to approximately its normal speed within one or two minutes. To meet these demands upon it, the author designed the governor shown in Fig. 195, and now well known as the "Richardson" Governor.[1]

Richardson Governor. There a central spindle *A* has a crosshead *F* formed at its top, by means of which the balls and stem are driven, two tooled surfaces, *G G*, in the

[1] Patent No. 14,753, 1885.

head of the sleeve serving for that purpose, and being used in preference to a wide surface, on account of the smaller friction. The arms, *H H*, are connected to the sleeve at points wide apart from the centre of revolution so that the balls being inclined inwards when at rest, may fly out in a practically staight line, *E E*. At the ends of the cranked arm, *H H*, are hard steel rollers moving with little friction on the true top

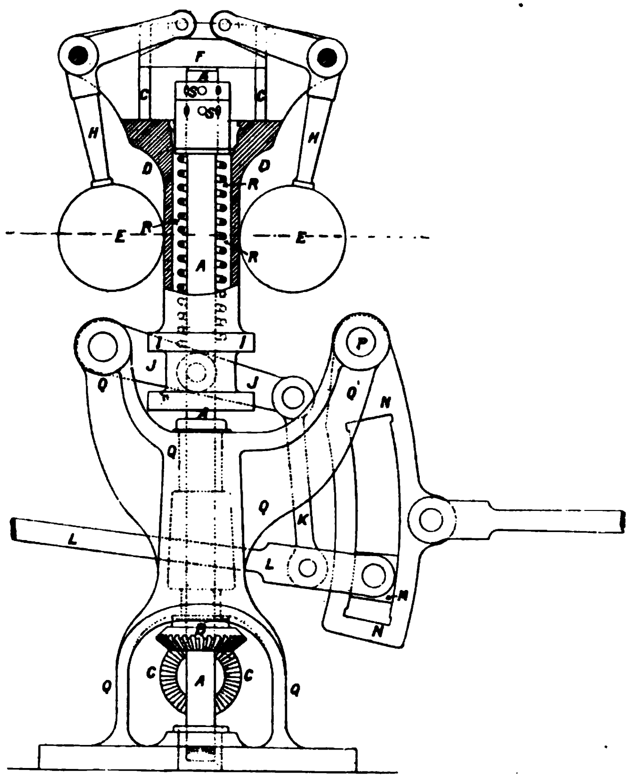


Fig. 195.

of the spindle *F*. As the balls expand, the arms fly out, the crank arm pressed on the spindle and lift the whole sleeve and balls together. This upward motion is resisted by a strong coiled spring *R* contained within the sleeve. The sleeve fits and slides upon the spindle at its lower end and upon the collar *S S* at the upper end. These collars form the abutment for the spring, the strength of which can be adjusted

within narrow limits by varying or turning the collars which are screwed on the spindle *H*, one serving as a lock nut to the other. The way in which motion is given to the radius link *N* through the governor fork *J* is so clearly shown that description is not needed. Such an arrangement permits the use of a long, powerful and flexible spring. The stability of the governor is very great; it can move the gear described with ease and rapidity, and can maintain the link in any position with perfect steadiness without the aid of a dash-pot. It is also very powerful for its size; one weighing only 53 lb. requires a force of 416 lb. to move it from its lowest to its highest position when at rest, and 200 lb. when in motion.

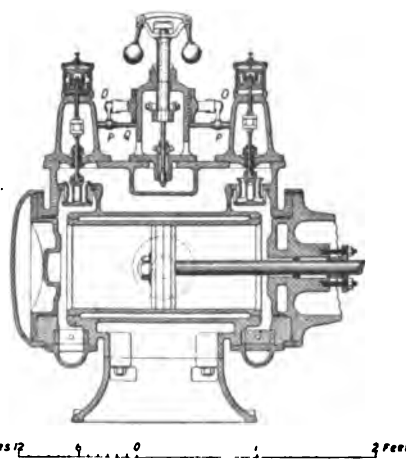


FIG. 196.

Only a very small governor of this kind is required to control a large engine when fitted with the drop valve expansion gear described in chapter VIII, page 148. A section of a cylinder fitted with this gear, and showing the governor in position, is shown in Fig. 196.

It is evident that Fig. 195 may be used to give motion to, and work gear and valves of any type which can be worked by other forms of governors.

CHAPTER XI

Electrical Regulation

THE conditions which obtain for regulating engines driving dynamos, are not quite the same as those for maintaining a constant speed in other machinery. If the electric current produced by the engine be utilized to control its speed, it then becomes possible either to run the engine at a practically uniform speed in connexion with a perfectly compounded dynamo, or to run it at a varying speed, maintaining either a constant difference of potential with a varying current, or a constant current with a varying difference of potential. Further, by utilizing the changes in the current to control the engine, it is not only possible, but quite easy, when the load is suddenly decreased, to effect what is impossible with a speed governor, viz. to shut off for a time the whole of the steam supply before the engine has had an opportunity of increasing its speed, and thus to deprive the engine of steam until its speed has fallen to just what is necessary for the diminished load, the steam supply being gradually increased until exactly sufficient for the load is supplied. Conversely, when the load is suddenly increased, and before the engine has had time to diminish its speed, the fullest supply of steam is given, and only then diminished to an amount corresponding with such increased load.

Motor Governor. In the following figures are shown a number of appliances which have been devised and tested by the author for the electrical regulation of steam engines.[1] Some of them, however, are shown as examples of what may be avoided, because, though apparently plausible in theory, they fail in practice. Fig. 197 shows the direct drive of a governor by electricity, which, though the first successful attempt to control the speed of an engine by the electric

[1] Patents No. 288, 1881; No. 941, 1882.

current, was an indirect method of attaining the result. The two small magnets under the valve were intended to act as weights to close it should the current be broken in any way, and thus stop the engine, these being of course held out of position by magnetism during the time the current was passing. In another form of governor, Figs. 198, 199 and 200, there is no solenoid in direct connexion with the valve, and only a small one E, moving a light lever, at the end of which are two platinum points. This lever and the fixed points A and B are so arranged that when the lever is in contact with the upper point, the small motor C is caused to revolve and by means of the crank lever F the valve is closed, and of course the reverse action takes place when the lever is in contact with the lower point. By this means it was thought that, as the solenoid at a certain intensity of current must

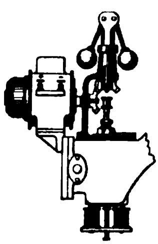


FIG. 197.

FIG. 198.

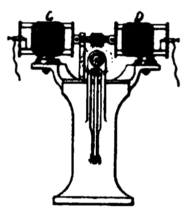


FIG. 199.

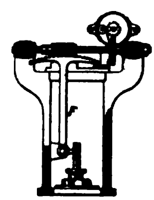


FIG. 200.

be maintained in one position, when this intensity was arrived at the lever would remain at rest. Careful tests, however, show that this is not so; however small the time occupied in making contact and setting the motion to work, it is enough to cause the engine to vary its speed too much, and the result is persistent hunting. This apparatus, therefore, must be considered a failure, though several modifications of it have since been introduced.

The Solenoid. Fig. 201 illustrates a very simple arrangement, in which the equilibrium valve A, having a spindle projecting through a stuffing box, is pressed upwards by the steam with a force due to its unit pressure multiplied by the area of the valve spindle, say 15 lb., and is held down by the short arm of the lever B, which has a ratio of 10 to 1, so that the force required is $1\frac{1}{2}$ lb. The electric current circulating in the two solenoids C holds up two iron cores weighing $10\frac{1}{2}$ lb.,

and a very slight variation in the current tends to cause a considerable alteration in the position of the cores, especially when they are placed at the position of maximum effect, as in Fig. 201. It is remarkable how slight a movement of the valve is required to effect the regulation.

Small Range of Valve Opening. Very careful experiments have shown that for an engine working up to 25 H.P. with equilibrium valves of 2 inches diameter, it required to be raised only 0·003 inch from its seat to run the engine at full speed unloaded; and it needed to be raised only 0·031 inch to run the engine at its full power—the variation or actual lift of valve being thus only 0·028 inch; and, as the ratio of the lever is 10 to 1, the amount of movement of the solenoid cores is only 0·28 inch. When a load is put upon the engine, however, the fall is always much more than this, and occurs instantaneously, and before the engine has had time to diminish its speed. The valve is thus opened for a few seconds three or four times as wide as in its normal position until the engine has received its full extra power; the solenoid then rises to nearly its former position. The reverse action takes place when the load is thrown off; the governor, simultaneously with the reduction of load, closes the valve entirely, and keeps it closed until the momentum is so far expended that the engine slightly slows, when the valve is gradually opened, just enough for the reduced load. The action is instantaneous and direct, and there is no hunting. This apparatus is suitable for comparatively small engines, say up to 50 H.P.

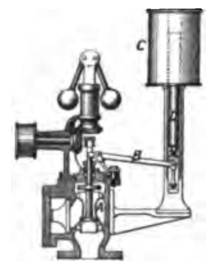


Fig. 201.

The means by which very large engines may be controlled by electricity will be shown later.

Automatic Stop. Figs. 202 and 203 show other forms of governor, with automatic arrangements for stopping the engine in case of failure of the electric current. In Fig. 202, which was the first form used, a heavy weight *A* is held up by an electric magnet *B* so long as the current is passing, but is released on the current stopping, when the lever *C* falling upon the valve spindle instantly closes the valve. The spring *D* and the weight *E* were for regulating the solenoid

to the exact quantity or intensity of current required, as was also the lever F, which, passing over the contacts G G G, disconnects one or two of the coils and puts them out of circuit. A simple way of stopping the engine is shown in Fig. 203,

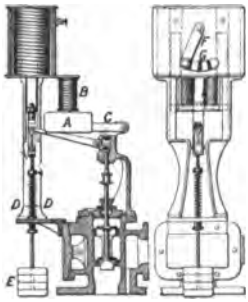


Fig. 202.

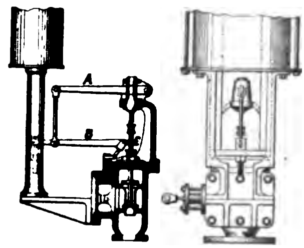


Fig. 203.

where the weight of the solenoid cores is used to close the valves on breakage of the current, acting through the lever A. While the solenoid is active, the regulation is effected through the lever B, A moving idly, but the instant the current is broken, the cores fall, the lever A presses upon the valve-stalk through the plunger C, and this is free to close; because the small end of the lever B works in a slot in the spindle with clearance upwards, all of its work being done and its only contact being on its lower edge.

In Fig. 202, F G shows a lever and a number of contact pieces, each of which is connected to a different set of coils on the solenoids, and is for the purpose of adjusting just the number of coils which are required in order to maintain the cores of the solenoids at the best working point.

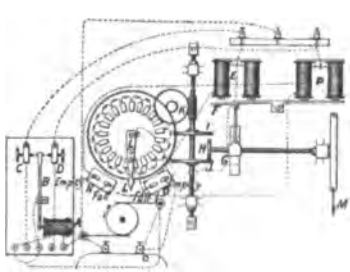


Fig. 204.

Relay. This arrangement was discarded later and replaced by a separate resistance box. In this box were a large number of German silver resistance coils, each one coupled up to a contact piece as shown at L, Fig. 204. The centre of the lever L is shown coupled to one of the dynamo

terminals O. The contact pieces are so connected together and to the other terminal of the dynamo, that any number of the resistance coils from none to the whole number can be put into the current. As the greater number are put in the current becomes weaker, the cores fall, more steam is admitted, and the engine increases in speed; the reverse takes place when the current is increased. Once the adjustment is made and the required number of volts or ampères obtained, then the further regulation can safely be left to the governor. The remainder of the apparatus shown in Fig. 204 was devised in order to make the adjustment of the lever L automatic.

The small wheel M was driven by a cord from the engine or dynamo as was most convenient, one of its shaft bearings was fixed, and the other had a small movement up or down, so that the small wheel H fixed on its end could give motion alternately to the discs I and J, thus causing the disc shaft to revolve either to the right or to the left, or to remain at rest when the wheel touched neither of the discs, a small spring keeping it in middle position when at rest. This small pulley is moved in the following manner—A is a small solenoid, the core of which is attached to the lower end of lever B, a spring pulling it back and the magnetism of the solenoid pulling it in. When the two forces balance each other, the lever B remains vertical. At the upper end of B are two platinum points, playing between the fixed points C and D. The small solenoid A is connected with the dynamo terminals. As the main current gets less it becomes weaker, the lever B inclines to the right, touches the point D, and excites the electro-magnet P. The armature is drawn up and the wheel H lowered on to the disc J, setting the vertical shaft in motion and moving the resistance lever L to put more coils in circuit. The current round governor solenoid is lessened, the cores fall, more steam is admitted, and the speed of the engine is increased until the required voltage is secured, when the solenoid A will release the lever B, restoring it to vertical position, when no further movement takes place, as the springs on either side of sliding beams G bring the pulley into a central position again. If from any cause the voltage were too high the reverse action takes place.

This apparatus worked perfectly and very delicately, the slightest change in voltage was felt instantaneously and re-

sponded to immediately, but in spite of that, it took some time, however little, to make a change, and being of the nature of a relay, it always dragged a little behind the needs of the engine, thus the control was not so good as when the adjustment of the resistance box was done by hand once for all, and the full control of the engine left to the electric regulation.

In the author's opinion, anything in the nature of a relay is detrimental to the proper working of an electrical regulator, though curiously, most of the inventors have introduced them. The only other electric regulator which has been at all largely used is, so far as the author is aware, that of Mr. D. W. Willans, described by him in his paper read before the Institute of Civil Engineers in March, 1885.[1] Mr. Willans describes his first governor in the following words:—

Willans' Electric Governor.

"In this governor the core *Q* is attracted by the coils of the solenoid *S*, and this attraction is balanced by the spiral spring *X*, from which the core is suspended by the rod *R*; on the rod *R* is a washer *Y*; against this washer a roller *A* presses. This roller is carried on the end of a differential lever *A B C*, the lever being prolonged in order to carry a balance weight *Z*, which can be regulated in such a manner that the roller *A* presses more or less on the washer *Y*. The point in the lever *C* is connected to the rod joining the piston in the relay cylinder *W* to the throttle valve *T*. To the point *B* is connected the rod of the piston valve *P*. If the current through the coils of the solenoid *S* increases, the core *Q* is drawn further into them, and the piston valve *P* is also moved down. The water under pressure is thus allowed to act on the under side of the water piston, entering the cylinder by way of the pipe *E*, the annular space between the two pistons of the piston valve placing this pipe in communication with the passage *K*. The water piston moves up, closing the throttle valve *T*, which is shown as a piston working in a perforated cylinder. The water above the water piston is discharged by the upper of the two pipes *F F*. As the water piston moves up, the point *C* in the differential moves up with it, and to a less degree the point *B* also, the motion continuing until the piston valve *P* is brought back to its original position, when the mechanism comes to

[1] *Proceedings of the Institute of Civil Engineers*, vol. lxxxi, part 111.

rest with the throttle valve in its new position, until some fresh disturbance in the circuit or in the steam pressure calls

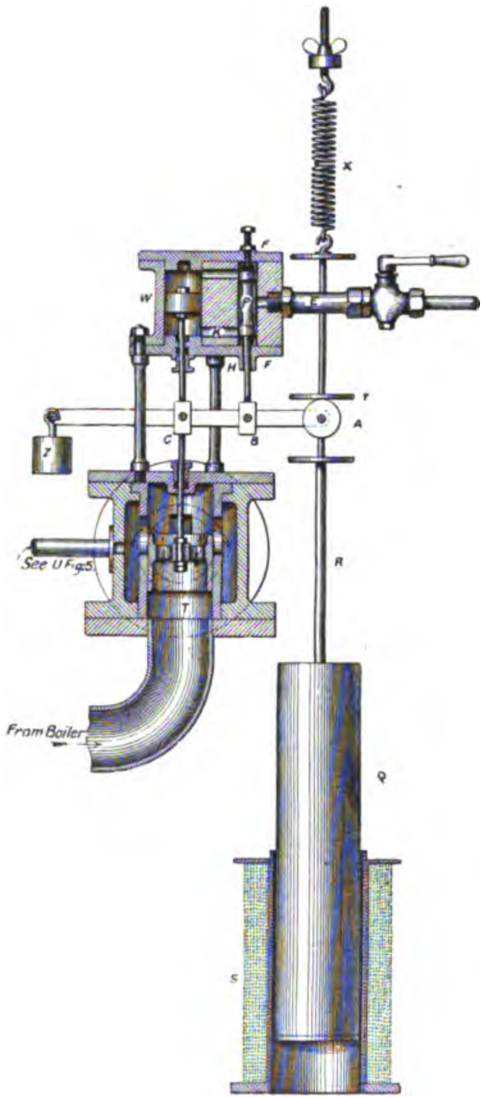


Fig. 205.

it again into action. The movement of the two points A and C in the lever is practically simultaneous, so that to the ob-

server the point B appears to be a fixed fulcrum on which the lever rocks, the movement necessary to open the passage being very small."

Mr. Willans goes on to say that "the principal difficulty at first encountered was that of 'hunting' and many devices were tried to get over it, such as dash-pots to prevent the core moving quickly, and cocks on the water-pipes to cause the relay piston to move slowly," . . . "but they introduced a worse evil in the slower action of the governor whenever a change in load necessitated a rapid alteration in the position of the throttle valve; the result of this slower action being a considerable temporary alteration in the intensity of the lights." "It was long," he continued, "before he realized the necessity for *absolutely simultaneous movement* of the regulating mechanism and the relay piston." "Eventually all dash-pots were abandoned, and the hunting was overcome by increasing the area of the passages between the piston valve and relay cylinder, so that the throttle valve could move as quickly as, and as nearly as possible in unison with, the core of the solenoid."

Fig. 206 shows the Willan governor eventually, as ordinarily employed, and is described by the inventor as follows :—

"In this type of governor the travel of the core is necessarily the same as that of the relay-piston, and it is not therefore suitable for any but small ranges of position in the regulating mechanism. Otherwise the arrangement is a most convenient one, for the throttle valve, water cylinder and solenoid are placed one above the other in the same axial line. The friction of joints and pins is avoided, the piston valve being connected directly to the core of the solenoid. No packing is required on the piston valve rod as it passes down the tube H which is high enough to prevent any splash of water. The piston valve in this type of governor works inside the hollow rod of the piston in the relay cylinder. The water enters the annular space between the two pistons of the piston valve by a port, E, in the side of the cylinder, and a recess, V, in the relay piston which is made deep on purpose. From this recess the fluid under pressure passes through a drilled hole into the interior of the hollow piston rod, the position of the hole being such that it is never covered by the valve. The passages leading from the piston valve to the two ends of the relay

cylinders are so arranged that the relay piston follows the motion of the piston valve, and comes to rest when it overtakes it. The passage which is covered and uncovered by

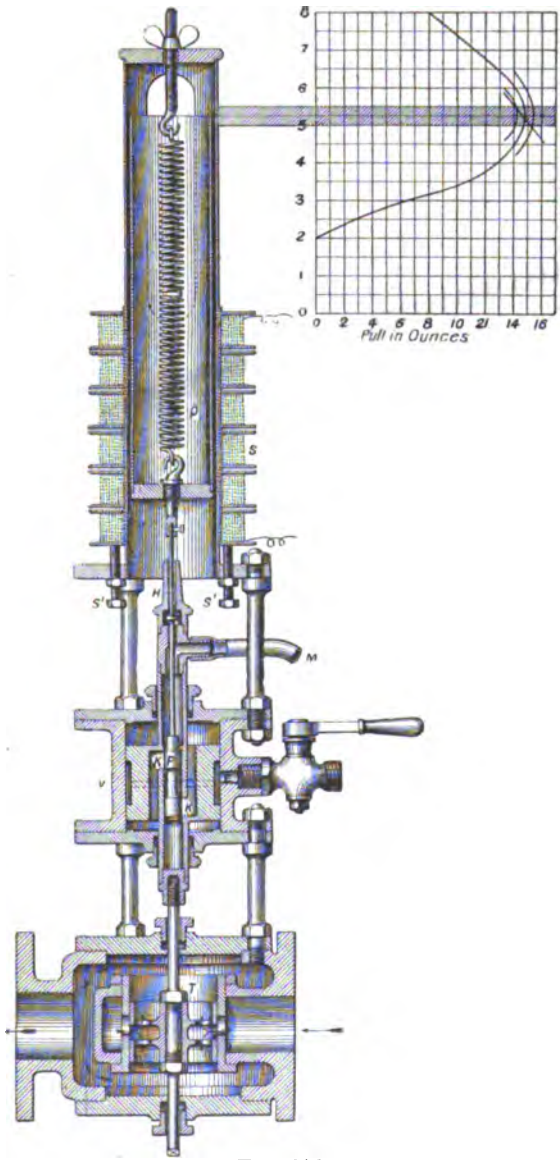


FIG. 206.

the upper piston of the piston valve leads to the bottom of the relay cylinder, and the one which is similarly covered and uncovered by the lower piston of the piston valve leads to the top. If the current through the coils of the solenoid *S* increases, the core *Q* is drawn further into them. The piston valve *P* moving down allows the water under pressure to pass through the pipe *E* to the upper side of the relay piston by the passage *K*. The relay piston then moves down until it overtakes the piston-valve *P* and closes the passage *K*. The water from the under side of the relay piston passes away above the upper piston of the piston valve by the passage *K*, and afterwards by the flexible pipe *M*. The core and relay piston move practically together, and it requires close observation to see that they are not rigidly connected. The throttle valve, which is in this case also of the piston type, is double, the ports admitting steam being formed in the side of the sleeve in which the double-throttle valve works.

"The governors are usually made with six coils wound with No. 30 wire. When these are coupled six in series they give the necessary pull with an electromotive force of 100 volts; when coupled three in series and two parallel they give the same pull with 50 volts; and when coupled two in series, three parallel, they give the same with 33 volts. The resistance of each coil is about 55 ohms, making 330 ohms when all are in series, and the current due to 100 volts is about 0·3 ampere. The electrical energy absorbed in the solenoid is thus 32 watts, or about ½ that of a twenty candle-power Swan lamp; such a solenoid is capable of controlling with ease the throttle-valve of an engine of 60 I.H.P.

"The method of winding the six coils separately is very convenient, as it renders the same governor suitable for any electromotive force between 30 and 100 volts, whilst at the same time the cooling surface is increased.

"The curve in at Fig. 206 shows the pull on the core of the solenoid when the current due to an electromotive force of 100 volts is passing through the coils. The pull is shown by the horizontal ordinates at each ½ inch of the stroke; the attractive force of the coils commences to be felt by the core just before the latter enters them; it reaches its maximum when the core is about half in and half out, and diminishes from that point until it disappears when the centres of the coils and of

the core coincide. The part of the travel of the core which is used by choice for regulation is near the apex of the curve. Here the pull is practically constant for a distance equal to the travel required by the throttle valve, and as the position of the apex of the curve does not differ appreciably with varying currents, its height and consequently the mean electromotive force to be maintained, can readily be determined by adjusting the mean pull of the spring by the wing nut at the top. The diagonal line cutting the curve represents the tension of the opposing spring. This diagonal line only coincides with the pull of the spring at one point in the travel of the core."

Defects of Relay. Extremely ingenious as are the above governors, they do not in the author's opinion satisfy the requirement of an electric regulator inasmuch as the action always takes place *after* the cause, when it is advisable that it should take place simultaneously with it; this is impossible with any system of relays, however good. Mr. Willans agrees with the author as to the requirements of a governor, but it was long," he states, "before he realized the necessity for absolutely simultaneous movement." This he evidently did not attain, because after all he speaks of an arrangement by which the throttle valve should move as "nearly as possible" in unison with the core of the solenoid. In the "Richardson" governor this is accomplished perfectly. When an engine is working under great changes of load with any mechanical or relay governors, as soon as the load is removed the engine runs faster, and then the governing mechanism comes into play and reduces it again to its proper speed, but with a *direct* electrical governor, at the same instant that the load is taken off there is an increase in intensity of the current, and the core of the solenoid rises and cuts off the steam *before* there has been any acceleration of speed. The slightest increase in current tends to lower the speed of engine generating it.

Momentum of Cores. So far from the weight of the core being a disadvantage as stated by Mr. Willans, the author finds with the direct action, that it is a decided advantage, the momentum due to their weight causing them in the first instance to shut off steam entirely for a short time, when the load is reduced admitting it again in a reduced quantity to suit the lesser need. Conversely, when a sudden great

load is added at the same moment a greater quantity of steam is admitted to meet the sudden change, this being reduced gradually to such a quantity as is required. With a directly connected solenoid and properly proportioned parts there is no tendency to hunt.

With engines of more than 100 H.P. a better method of steam control is required than an equilibrium throttle valve, and after many experiments a form of electrical

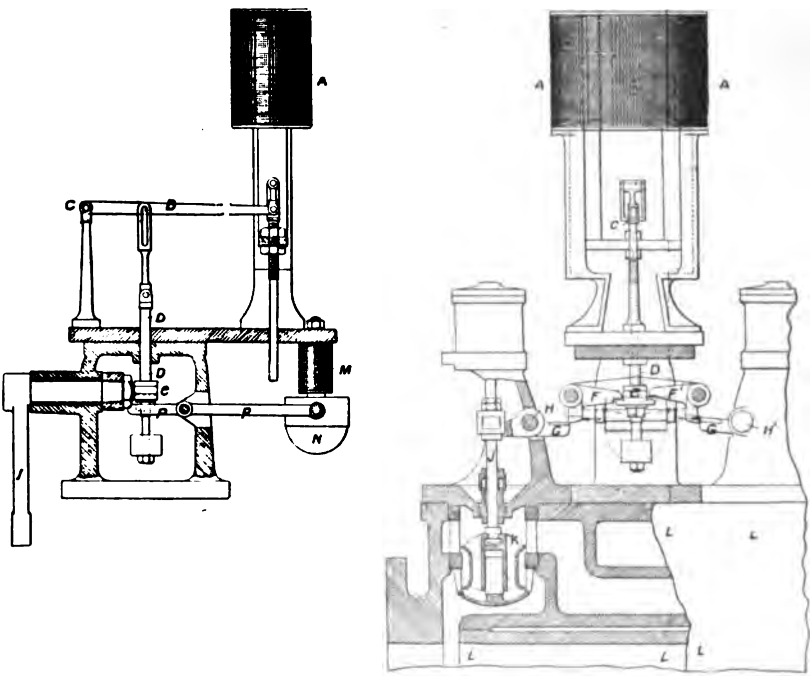


Fig. 207.

control of the valve gear of very large engines was designed in conjunction with Mr. Ralph Nevile.[1] This was first applied to an engine fitted with a Proel expansion gear and is shown in the specification drawing (Fig. 207) and clearly explained in the letterpress as follows :—

Richardson-Nevile Gear.

"The way in which the solenoid is made to act upon the trip or releasing gear will be clearly understood by reference to the annexed drawings in which Fig. 207 are side and end

[1] Richardson and Nevile patent, No. 5,025, 1886.

elevations respectively of the improved apparatus for controlling the speed of a steam engine. *A* is a solenoid which may be double as shown, or single; it is connected by the lever *B*, whose fulcrum is at *C* to the rod *D* by means of which it raises the collar *e* upon which the tripper levers *F F′* rest. The short arms of the tripper levers *F*, *F′*, raise alternately the levers *G* and *G′* (whose fulcrums are at *H* and *H′* respectively) by the means of the arm *I* which is connected to the engine excentric or by means of any well-known or suitable mechanism driven from the engine. The levers *G* and *G′* operate upon their valve rods *J* to admit steam through a valve *K* at alternate ends of the cylinder *L*.

"As the collar *C* is raised by the solenoid the short arms of the levers *F* and *F′* are drawn inwards in the direction of the arrows, and thus the levers *G* and *G′* are raised earlier and the cut-off of steam takes place at an earlier part of the stroke. *M* is an electro magnet in the same electric circuit with the solenoid *A*, and *N* a weight forming the armature to the same so that if the electric circuit be accidentally broken at any part thereof, the weight *N* falls and by means of the lever *P* raises the trippers *F* and *F′* and cuts off the steam."

This arrangement gave exceedingly good results, and no trouble whatever, as was shown by the following communication to the Institute during the discussion of the author's paper[1]:—

"Mr. F. W. E. Jones referred in tones of praise to the electrical method of regulation, as far as he had had experience of it, at Leamington Spa. He had under his control there two compound condensing-engines, each capable of developing 250 H.P., which were fitted with electrical governors practically the same in arrangement as shown on Fig. 207. Their solenoids were excited by either of a pair of dynamos driven by each engine. The conditions of working made it essential that the speed of the engines should be capable of variation, in steps equivalent to 1 volt at a time, and in accordance with the variations of load occurring while the engines were at work. This was effected by an adjustable resistance placed on the instrument table and under the control of the attendant. The regulation was perfect in every respect, and any desired

[1] Minutes of *Proceedings of the Institute of Civil Engineers*, vol. cxx, 1894, part II.

increase or decrease of electromotive force was immediately obtainable either in steps of 1 volt or more at a time, according to the plates inserted or taken out of the resistance. It seemed to him that it would be equally simple to arrange for the addition or otherwise of any fraction of a volt if requisite, or, by a modification, for a constant speed irrespective of the load. He noticed that no case had come to the knowledge of the author in which the action of a safety appliance or a mechanical governor had been needed. In this station, however, through a fault which developed in the winding of one of the solenoids, the mechanical governors had been in one instance called upon to prevent the engine running away; with solenoids as the author suggested upon a vulcanite, instead of upon a brass bobbin, this failure would not have occurred. In his opinion where absolute uniformity of speed under varying loads was necessary an electrically-controlled engine would be found more satisfactory than one governed by any centrifugal arrangement."

During the discussion of the author's paper the following statement was made by Professor Unwin in the course of his remarks :—

Professor Unwin's Dictum.

"With regard to the special problem in governing the speed of an engine, there were two sets of causes of fluctuations of the speed. There were causes which could be classed as periodic, occurring within the the time of a revolution, and causes which were aperiodic. So far as the periodic causes of variation of speed were concerned, a governor could have no action whatever. The governor could not regulate the engine within the time of a revolution, and it was the fly-wheel alone that must be depended upon to neutralize to a large extent those periodic fluctuations of effort, and as far as possible to remove them from having any effect upon the governor. They resulted, of course, from the variation of the steam-pressure in the stroke, and from the variation in position and velocity of the working parts of the engine. With regard to the aperiodic causes of fluctuation of speed, the fly-wheel also had a very important function. It could not prevent the fluctuations of speed due to those, but it could make the variations so gradual that it was possible for a governor to control them. He did not believe that any consideration of governors was valid if

it did not proceed with the simultaneous examination of the action of the fly-wheel and of the governor. As to the governor itself, it was troubled first by the circumstance that it could only have effect on the regulation of the engine during the period of admission of steam, which was only a fraction of the revolution, and next, that in proportion as it was made sensitive, and capable of sensitively regulating the speed, its power of overcoming the frictions and so on, which could not be altogether avoided in any form of mechanism, was reduced. There was one type of valve-gear which could be extremely well controlled by any good form of governor. Any form of trip valve-gear could be controlled by a governor, so as to produce very considerable accuracy of speed; but in regard to any other form of valve-gear, in which the governor must exert more or less considerable effort in effecting the regulation, he thought the problem was one of great complexity and intricacy. He would only say that apart from engines using trip valve-gears, he believed the real solution of the problem of governing accurately lay in the use of the relay. If the governor be regarded as a pure thinking machine, and threw all the work of governing on some relay, there was a possibility of securing any accuracy of speed that might be desired. He might be told at once that relays had been tried over and over again, and that they had one universal fault of hunting. That was perfectly true of a great many relays that had been used. The relay had been used in a great many cases without any insight into one of the conditions absolutely necessary if relays were used at all. In most forms of relay regulation which had been used, the governor started the relay, and the relay went on acting without any control till, by the alteration of the position of the governor, it was thrown into action in an opposite direction. It was impossible to avoid an extreme form of hunting with a relay governor of that kind. If a relay were used at all, it should be used much in the form in which it was used in the stearing-gear of a ship, where the captain turned a small wheel which gave the signal to the relay to get into action; it also told the relay to do so much and no more. The relay came to a stop when it had turned the rudder over to a certain small extent. The captain then considered whether enough had been done; and if not, he gave a further signal, and the relay did a little

more. It was only when the relay was used in that form that it could properly control the speed. Governors of that type had been used—hydraulic and ratchet relay governors—to control large sluice-valves of turbines with remarkable accuracy. These turbines were used for driving dynamos in electric-lighting installations, where the load was necessarily varied. The dynamos ran with great regularity of speed, and the whole secret was, first, that the force required to do the regulation was thrown entirely upon the relay, and, secondly, that the relay was of the stearing-gear type. It had been stated that if a relay governor of the stearing-gear type were used it could not be isochronous. In general it was not desirable that a governor should be isochronous, but it was possible that a relay governor could be virtually so if necessary. The automatic action of the relay in coming to rest after being started by the governor had been made also to readjust the position of the point of suspension of the governor arms. Thus, if the speed of the motor increased, the governor height diminished, and the relay was started. As the relay acted, it automatically came to rest again, and simultaneously corrected the error of governor height."

Simultaneous Action.

With most of what is stated above the author is in agreement, and with most governors, and with all relays, it is true, that "the governor could not regulate the speed of an engine within the time of one revolution"; of course the fly-wheel must be depended upon to give an average speed during one revolution, but with the electric governor shown and described in the author's paper, it is possible to materially change the speed within the space of one revolution. In the case of governing with a throttle valve *all* the steam is shut off for a time simultaneously with the reduction of load, *all* the power then depends upon the momentum of the fly-wheels which must decrease in speed to some extent immediately.

In the case of the trip gear where there are two cylinders, there are four times in a revolution when steam is admitted, and the speed of the fly-wheel depends upon its receiving these four impulses. The electric regulator can deprive it of three out of the four "within the time of one revolution." It is from the fact that the direct electric regulator acts upon the steam simultaneously with the change in load, and therefore

before there can have been any acceleration or diminution in speed, however slight, that it has such a manifest superiority over any other form of governing.

The vast improvements which of late years have been made in dynamos by which they can be compensated to such a degree by their windings, that a slight change of speed does not materially affect the voltage, makes the necessity of electric regulation less pressing, otherwise it would have been much more largely used than it is; there is, nevertheless, still a wide field for it in connexion with tramway work, where the changes in load are extremely great and frequent.

Professor Unwin's statement is true at any rate for large engines that either a relay must be used or trip gear adopted. That the governor should be regarded as a "thinking machine," throwing all the work of governing on a relay, would only be true were it impossible for the governor to think and act at the same moment. This is most accurately accomplished in the latest form of governor shown in Fig. 207A. This governor acts in connexion with the cut-off gear shown in Fig. 196, shown on p. 198.

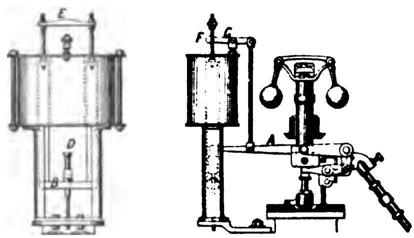


FIG. 207A.

Trip Gear. In this figure, *A* is the governing lever regulating the admission of steam by releasing the tripper as previously described, the vertical motion of the lever is only about half an inch. The outer end of the lever rests upon the stud *D* which is attached to the cross-bar connecting the two solenoid cores. At this point *D* there is a slight magnetic attraction keeping the governor lever and the stud in perfect though quite free and easy contact, so that as the solenoid cores rise and fall with any variation in the strength of the electric current, the governor lever rises and falls with them, and cuts off earlier or later as is needed.

The upper ends of the solenoids are connected to the cross-bar *E*, which forms part of the safety apparatus.

In case of the breakage of a connecting wire or any other interruption in the supply of current, the cores would immediately fall, the cross-bar *E* would then come into contact with the end *F* of the lever *C*, the opposite end of which would be lifted up carrying with it the governing lever *H*, and thus entirely cutting off the supply of steam and stopping the engine.

In cases where the engine may be used for other purposes as well as the generation of electricity, a centrifugal governor is used which only comes into action if the speed is increased 2 per cent. beyond the normal; this is also shown in Fig. 207A. As there is no upper collar to governor slide, the centrifugal governor has no influence whatever upon the governing so long as electricity is in action.

For this safety apparatus and for much help in carrying out very many experiments in connexion with the governor, the author was much indebted to Mr. Ralph Nevile, of Wellingore Hall, whose name is joined with his in the patent.

The subjoined report by Professor Sylvanus Thompson on the working of one of the first made electric governors will be of interest as showing amongst other things how a steam engine can be made to reduce its speed in proportion to its load and how "simultaneous" is the action of the governor with the change in load.

"The governor was attached to a high pressure double-cylinder fixed engine of 10 H.P. (nominal) working under 70 lb. pressure of steam and with ½ stroke cut-off. The governor controlled the supply by an equilibrium valve. The actual output of power was about 25·5 H.P. when the engine was doing its greatest amount of work. The machine driven by the engine was an ordinary 16 light brush machine, and at the time of my examination was actually driving 17 lamps. These burned very steadily except one lamp which was evidently out of adjustment, and which at one time refused to light, but burned steadily when lighted. The dynamo was driven by a 7-inch belt direct from the engine. During the tests which I applied the speed was read by a Buss's tachymeter, the strength of the current by an Ayrton and Perry's ampèremeter.

"The governor which operated the supply valve consisted

of a double solenoid, placed in the main current of the electric wire and of a double iron core which was drawn up by the attraction of the current against the reaction of a spring and a counterpoise weight. This arrangement acted on the valve by means of a lever which opened the valve when the current diminished in power, and closed it when the current lifted the weight beyond its normal position. It therefore tended to maintain a constant strength of current irrespective of the velocity of driving.

"My tests were directed mainly to two points. *Firstly*, to ascertain whether the current was maintained in reality at a constant strength when different numbers of lights were switched into the circuit. *Secondly*, to ascertain the rapidity of action of the governor in order to compare it with centrifugal governors.

"The following table shows the strength of current and the speed observed at various times when different numbers of lamps were burning :—

TABLE I.

Lamps.	Revolutions.	Ampères of Current.
17	146	10·2
17	144	10·2
17	143	10·1
17	137	10·2
17	133	10·2
17	119	10·2
16	133	9·9
16	132	10·1
16	129	10·2
11	107	10·1
11	101	10·0
11	96	10·2
11	92	10·3
11	90	10·3
11	89	10·3
11	85	10·5
5	70	10·0
5	60	10·5
0	30	10·4–10·6 } un-
0	24	11·0–11·2 } steady.[1]

[1] The unsteadiness observed was due to the separate electric impulses, as the separate parts of the commutator moved past the brushes of the dynamo, each impulse at this low speed producing a movement of the pointer of the ampèremeter.

"From the foregoing table it appears that at all speeds from 25 up to 146 revolutions per minute and with any number of lamps from none up to 17 alight, the current was practically maintained at a constant value in the most efficient manner.

"The values collected in this table are from a large number of experiments, and are placed not in the order in which they were made, but in the order of the speeds. The observations are instructive in several points. The first set—made with 17 lamps—show how the speed 119 at the moment when all 17 lamps were fairly kindled, rose gradually (following the order of numbers to the top of the list) as the resistance of the arcs increased to 146 revolutions in about 15 minutes, the current all the while keeping *within* 1 *per cent.* of its constant value.

"The extreme difference between the strengths of the current when 5 lamps were lighted and when 17 were lighted was only 6 per cent. of the whole constant value, and even when all the lights were extinguished the change in the strength of the current was only 12 per cent., the engine continuing just to crawl round and so keep the magnets of the dynamo charged.

"The time tests showed the following results. The table gives the numbers of lamps turned out and left lighted, the momentary fluctuations observed in the current and the speed, and the time taken to settle down to a steady condition.

TABLE II.

Lamps.			Fluctuations of of Current.	Fluctuations in Speed.	Time required to Settle Down (seconds).
Alight at First.	Switched Off.	Left Lighted.			
5	5	0	10–15–10·4	—	8
16	5	11	10·2–13–9–10·1	—	under 4
17	6	11	10·2–14–10	138–107	4
16	11	5	10·2–17–9–10·5	—	9
17	12	5	—	140–60	11
17	17	0	—	133–24	14
17	Circuit broken purposely.			140– 0	55

"When lamps were suddenly turned on similar fluctuations were observed but never so great, the current falling a little

at first, then steadily returning in 10 or 12 seconds to its normal value. The speed usually took several minutes to recover, as the lamps newly lighted went on for some time increasing the resistance of their arcs, which though all short at first lengthened as the carbons burned away a little. In the last experiment set down in Table II the circuit was suddenly opened while all the lights were burning. Had the engine had only an ordinary centrifugal governor the sudden removal of the load from the driving belt would have caused the engine to run away. As it was, all that happened was, first, an *instantaneous* drop of the dead-weight of the electric governor causing, secondly, an absolute cut-off of the steam within the quarter of a second of time, the engine coming to rest in 35 seconds. The momentum of the fly-wheel and other moving parts could not of course be reduced in much less time to zero. No centrifugal governor could have so instantaneously cut off the steam; it would not have acted until the engine began to race. With the electric governor the steam was cut off before racing could even begin. In my opinion the electric governor is not only more rapid and more sensitive than the ordinary centrifugal governor, but it is far more reliable. It cannot possibly get out of order, needs no attention and no lubrication, and does not require any driving strap or gearing beyond the lever of the valve. It can be adapted according to the circumstances of the system of electric distribution to which it is applied to maintain either a constant current or a constant electromotive force. It can also be adapted to control the actual output of horse-power in the electric circuit at a constant figure.

"In addition to these points the electric governor will prove of value from an economical point of view, since it cuts down the consumption of steam to the actual demand made upon the supply in the electric circuit.

"In view of the so-called automatic dynamo electric machines which have their coils partly series-wound and partly shunt-wound or wound in some equivalent manner, I am clearly of opinion that whilst such machines are or may be truly self-regulating under the condition of a constant speed of driving, the condition so presumed cannot in practice be maintained without some contrivance of the nature of an electric governor on the driving steam engine, and the gover-

nor is needed to preserve the engine as well as the dynamo-machine itself from suffering injury when any large number of lamps are suddenly turned on or off. A sudden breakage of the whole circuit might be disastrous to one of these so-called self-regulating dynamo-machines unless an electric governor were also attached to the steam engine to cut off the steam in the manner in which the steam is instantaneously cut off by the Richardson electric governor.

"(Signed) SILVANUS P. THOMPSON."

"*March* 7, 1883."

CHAPTER XII

Condensers

THERE are two principal purposes for which condensers are used in connexion with steam engines; one of these is for the reduction of pressure from that of the atmosphere to about 14 lb. below it thus causing an increase of power, and the other is for the recovery in the form of water, of the steam which has been used to work the engine.

Varieties. There are many kinds of condensers, but all may be included in one of the following, viz. Jet condensers, Surface condensers, Air condensers, Evaporative condensers, and Ejector condensers.

Injection. We will deal first with the most common type, viz. injection or Jet condensers, and we will see how they tend to reduce pressure and increase power. Each side of the piston of an ordinary steam engine is alternately steam and exhaust. From any pressure on the steam side of a piston we have to deduct that on the exhaust side in order to get the net pressure which does the useful work. As we have shown in previous chapters, the ordinary pressure of the atmosphere is about 15 lb. on the square inch and about 14 of this we can remove by the use of a condenser, when the exhaust steam is made to discharge into it, instead of into the atmosphere. Assuming then that we have an average pressure on the steam side of 28 lb. above the atmosphere in the low pressure cylinder of a compound engine, by discharging the steam into a condenser, its pressure can be reduced to 14 lb. below the atmosphere, and thus its effective pressure raised to $28+14=42$ lb., an increase of 50 per cent.[1] This is a gross, not a net increase.

[1] It is important that the student should have a clear idea of percentages and how they are ascertained.

An original amount such as the horse-power of an engine (or the capital of a man) may be theoretically increased to any percentage or

In order to condense the steam we have to inject cold water into the condensing chambers in considerable quantities, and this water and the steam bring a quantity of air with them, estimated at from 5 per cent. to 6 per cent., which would soon accumulate and destroy the vacuum ; this water and air have to be pumped out at each stroke, and the power required for doing this has to be deducted from the gross gain.

Air Pump.

Thus supposing our low pressure piston has an area of 1,000 inches, and is moving at the rate of 600 feet per minute with an average pressure above atmosphere of 28 lb., then $\frac{1,000 \times 28 \times 600}{33,000} = 509$ H.P. If instead of this we have 42 lb. the $\frac{1,000 \times 42 \times 600}{33,000} = 763$ H.P. (a gain of 254 H.P.). If now our air pump piston is moving at the rate of 200 feet per minute and has an area of 500 inches working against a net resistance of 14 lb. per square inch, we have $\frac{500 \times 14 \times 200}{33,000} =$ 42 H.P., and it would require another 8 H.P. for friction of its parts, which makes a total of 50 H.P. which subtracted from the 254 gained, gives us a net gain of 204 H.P., a very valuable

amount. Thus an engine giving originally 10 H.P. may by certain alterations be made to give out 5 H.P. more. This addition is manifestly an increase of 50 per cent. as 5 is the half of 10, but the figures are not always so simple. The formula is as follows :—" Divide the original sum by that added to it, and then divide 100 by the answer. Thus $10 \div 5 = 2$ and $100 \div 2 = 50$ per cent. increase. If the increase in power were 25 instead of 5 H.P., then $10 \div 25 = \cdot 4$ and $100 \div \cdot 4 = 250$ per cent. increase and so with irregular amounts.

When, however, we want to express any proportion of a given amount, such as a saving in fuel (or expenses) it can evidently never be more than the whole amount, or 100 per cent. and is generally less. Thus, we formerly used 10 tons of coal a week, now we only use 5 tons, manifestly a saving of $\frac{1}{2}$ or 50 per cent. This is worked as follows :— Multiply the smaller amount by 100 and divide the product by the larger amount. Thus $5 \times 100 = 500$, which divided by $10 = 50$ per cent. The statement is expressed as follows : $\frac{100 \times 5}{10} = 50$ per cent. ; if the saving were 10 tons out of 15 then $\frac{100 \times 10}{15} = 66 \cdot 6$ per cent. ; if the saving were 14 tons out of 15 then $\frac{100 \times 14}{15} = 93 \cdot 3$ per cent. Thus an increase of power (or capital) can be to any percentage, but a proportion of saving in fuel (or expenses) can never be more than, seldom is so much as, 100 per cent.

increase to the power of an engine. Assuming, as we fairly may, that the high pressure cylinder gave an equal amount of power with the low, then we should have the power of the engine increased from 1,018 to 1,222, or an increase of 22 per cent., and we may take roughly 20 per cent. as about the gain that can be obtained by condensing with a compound engine.

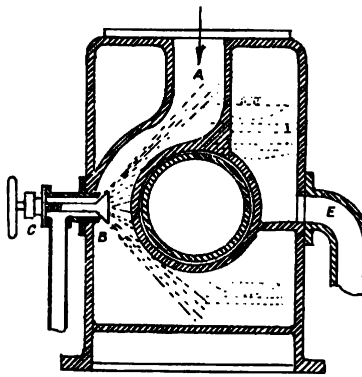


FIG. 208.

Water required. A jet-condenser cannot be used without water, and a considerable quantity is required, not less than 30 times the amount required for the feed when the temperature is 50°; if it be colder, then less will do, but for warm water as much as 50 times the feed may be required.

For comparatively small engines the air pump is generally worked by a prolongation of the piston rod. In such cases it is advisable (if there be room) for the condenser to be fixed so far away from the cylinder that the cold rod of the air pump never enters the hot steam cylinder.

There are many varieties of horizontal condensers and air pumps, a very efficient form being shown in Fig. 208, which is a transverse section, and Fig. 209, which is a longitudinal section of condenser and air pump.

The steam enters preferably at the top, as shown by arrows; by so doing the risk of returning water to the cylinder is avoided. Entering the condensing chamber at *A*, the steam meets with a very effective jet *B*. The injection is frequently, even

Pipe Injection. usually, in the form of a pipe with a number of small holes drilled into it, or a long narrow slit or slits, and these act very well if they are correctly proportioned to the quantity and pressure of the water coming through them, and above all, if the water is clean. Not infrequently, however, condensing water, especially that taken from rivers in towns, contains a good deal of impurity, and this tends to stop up the holes or slits in the injection pipes, and thus reduces the quantity of water injected. A regulating cock as at *C* is always

put on, but with such an injection pipe as described above, the only thing the cock can do is to regulate the admission of water, and, if the holes are stopped up, though the cock be full open, no water or only a meagre supply will be injected;

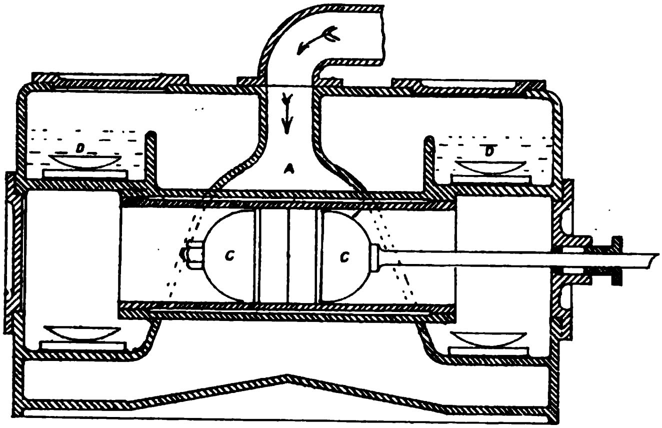


FIG. 209.

further, if the holes in the pipe are too large or too numerous, the injected water will dribble in, instead of being squirted in as it should be.

Conical Valve. With the conical valve shown at *B* (Fig. 208), the water is delivered in a thin continuous cone, filling the condensing chamber and exposing the largest possible surface to the entering steam. The steam is thus rapidly condensed. When water is delivered in this way a smaller quantity will produce a satisfactory result than when injected improperly, and in not a few instances the author has made a very material reduction in the quantity of injection water used by an engine by such an improvement of the jet, and, at the same time, not only produced a better vacuum, but left the water at a higher temperature in the hot well.

Another great advantage which the conical valve has over a pipe with holes in it, is the facility with which it can be cleaned. When it becomes clogged with dirty water, it is merely necessary once a day or more, just to open the valve wide when there is a good vacuum in the condenser, when the rush of water will speedily wash and cleanse the valve, after which it is easily adjusted again to the required quantity.

In this air pump there are eight valves, two suction, and two delivery at each end; they are readily accessible by taking off the cover at each end; the method of fixing the seats which are gun-metal gridirons, is clearly shown at *F* in Fig. 210.

Air Pump Valves. It is highly important that the delivery valve should always be perfectly sealed against a leakage, hence they work in open chambers containing at least three inches of water above them. The water overflow pipe to hot well is shown at *E* (Fig. 208). A pipe of about equal size to the injection supply should be fixed from the upper part of delivery chamber to carry off the air; it is more easily got rid of in that manner than if compelled to go with the overflow water. It is the best practice to line the pump cylinder

Piston. with gun-metal, and use gun-metal piston rings; they work better in water than iron and are subject to less injury. *C C* are blocks of wood, preferably willow; these serve three purposes: one is to fill up some of the clearance space, another to partially float the piston, and thirdly to form hemispherical ends to the piston so as to obviate or materially mitigate the shock of the water which is always greater in a pump only partially filled with water, than when it is quite full.

Such a condenser as shown in the two Figures, 208 and 209, works well and maintains a good vacuum, but for very large engines working with a piston speed of over 500 feet per minute, a vertical air pump of larger diameter and shorter stroke is preferable; such an one is shown in Fig. 210. Such a condenser is fixed well below the engine room floor, and when used in connexion with a horizontal engine is fixed generally behind the cylinder and worked from the piston tail rod. The tail rod has to work in a slide provided for it with a crosshead at the end which crosshead is connected by two short connecting rods with the upper end of the long arm of the cranked lever *A*.

When there is not available length in the engine room the condenser and air pump may be fixed in a chamber between the cylinder and crank shaft, as in Fig. 210A. In that case the long arm of lever *A* is coupled by two short connecting rods to the crosshead of the engine. The rocking lever is sometimes made of cast iron, but two steel plates separated by distance pieces through which bolts pass, are to be preferred. A very large pin with its surface hardened should be used for the

joint within the plunger. If properly proportioned and well oiled, it will work for many years without needing adjustment or repairs. If the lever arms be made three to one as shown then a piston speed of 600 feet per minute is reduced to 200 feet for the air pumps. It will be noticed that there are only two valves, both annular disc valves, working on gun-metal gridiron seats. In the down stroke of the piston the foot

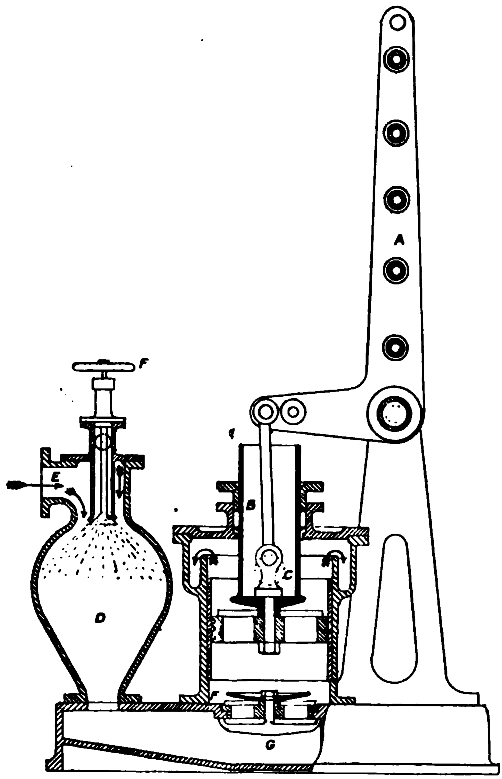


Fig. 210.

valve *F* closes, the piston valve opens, the water and air passing freely through to its upper surface. At the same time the trunk *B* comes down and its volume displaces an equal volume of air and water. Its volume being equal to about half that of the pump cylinder, half the capacity of the cylinder is delivered in the direction of the arrows through the discharge opening *C*, shown in dotted lines. This opening may

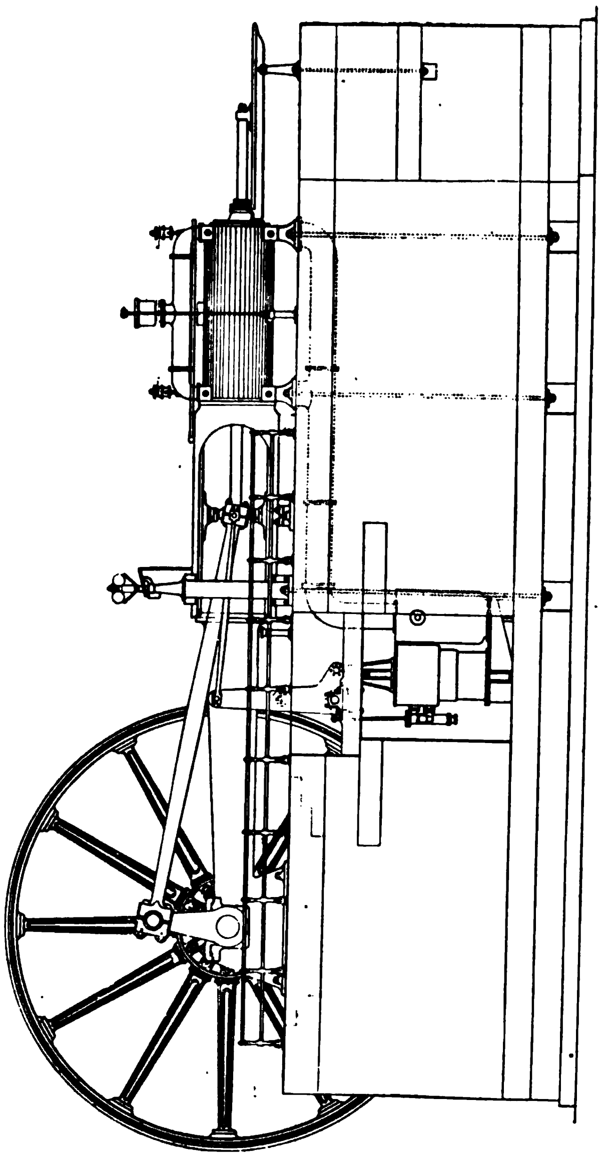


FIG. 210A.

be put anywhere in the circumference of the delivery chamber as is found most convenient.

In the up stroke the discharge is again half the cylinder capacity, viz. the annular space round the trunk, while a cylinder full is drawn in through the foot valve *S*. In this way with a single piston there is a continuous delivery all in the same direction, which is a decided advantage. This pump works very easily and gently, and in small sizes the author has worked them up to 600 double strokes a minute without shock or jar, the stroke in that case being only 3 inches, equal to 300 feet a minute.

Condensing Chamber. There are no head valves shown in Fig. 210 nor are any needed when the water flows away freely at or below the level of air pump, but when it has to be delivered at a higher level an extra head valve is needed.

The form of condensing chamber which the author has found to give the best results is that shown at *D* in Fig. 210. Here again a conical injection valve is used, and it will be seen that the steam entering at *E* in direction of arrows, must all pass the cone of injection water and be thoroughly condensed, such a thin cone of water offering the largest surface in proportion to its volume. Thus the best use is made of the injection water. The handwheel *F* is generally connected by a vertical spindle to a lever or wheel supported by a standard above the engine house floor. In this way the injection water can be regulated from that place.

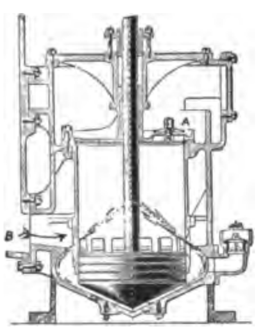


FIG. 211.

Edwards' Air Pump. Fig. 211 shows another form of air pump known as "Edwards' patent," and it has many good features and has been very largely used during the last few years. Amongst the advantages claimed for it are the following:—There is no foot valve and no bucket valve, thus all risk and trouble due to a failure of either of these is avoided. The delivery valve *A* is very accessible, and if there be no head on the delivery, this valve can be examined without stopping the pump. Further, that owing to the conical shape of base of the bucket working against a corresponding conical depression in the base of the pump, there is a total absence of shock, the

water delivered being silently projected into the working barrel.

The air and water enters at *B*, and while the bucket is a little removed from the end of its downward stroke, the water falls freely to the bottom while the air rises above the bucket. As the bucket descends further the water is compressed, and rising at each side, is forced through the series of holes shown in the working barrel above the bucket. As the piston or bucket rises, the holes are closed by its body, and the air and water are then delivered through the valve *A* into the overflow cavity.

It is not considered advisable to work the pump with a greater head of water than 4 inches above the delivery. Owing to the very small clearance, this makes an exceedingly good air pump. Unlike the pump shown at Fig. 210, the Edwards pump is only single acting. Except for very small installations, they are generally worked in pairs, or groups of three worked from a common crank shaft.

Surface Condensers. When the condensing water contains salt or other impurities so largely that it is unfit to be mixed with the feed, an injection condenser cannot be used, and means have to be adopted to secure condensation without such admixture. The plan generally adopted is to deliver the exhaust steam into a chamber in which are a great number of small tubes, within which tubes cold water is circulated in order to keep them cool. The steam is condensed upon the surface of these cold tubes, and forming water, falls to the bottom of the chamber, whence it is removed by a pump. Such an apparatus is called a surface condenser, and an ordinary form of it is shown in Fig. 212. It consists of a nest of tubes contained within a casing *A*. In small sizes these casings are made of cast-iron, and in marine engines are often formed of irregular shape within the engine framing. That shown in the illustration is made of steel plate and has a strong angle riveted on it at each end; the pipe connexions also, *J* and *K*, are riveted on. The angles at the ends are truly turned up and brass plates *B B* are bolted on at each end by collar studs. These plates need to be strong and tough, and should be made of a mixture of about 62 per cent. copper, 37 per cent. zinc, and 1 per cent. tin. Beyond the plate are the castings *F* and *D C*, the latter divided by the horizontal mid-feather *E*. The ends are enclosed by the covers *I* and *M*. The end castings and covers are shown in section, also one-half of the end view (Fig.

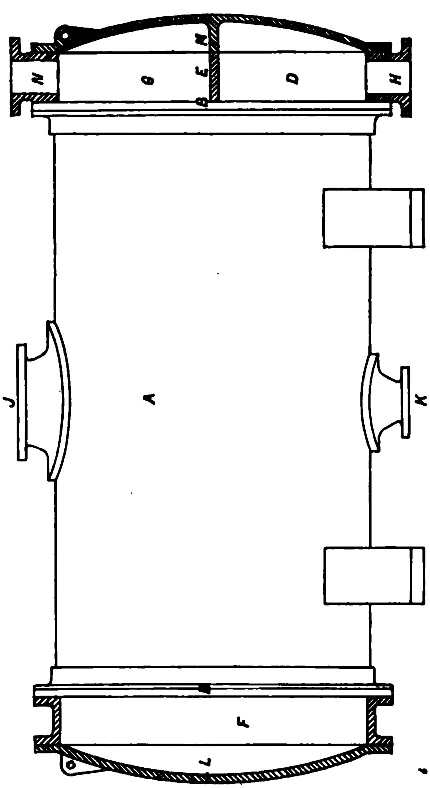


Fig. 213.

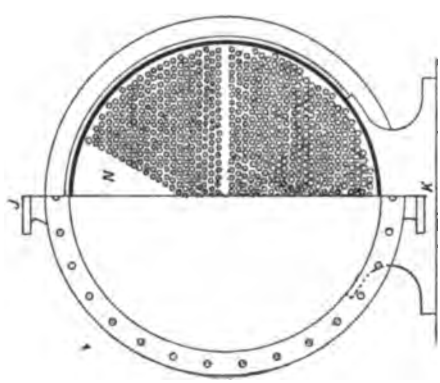


Fig. 212.

212.) The tubes, of from $\frac{5}{8}$ to $\frac{3}{4}$ of an inch diameter, are placed as close together as they can be to allow of the packing round them and about $\frac{3}{16}$ of an inch of solid brass. These are not more than $1\frac{1}{2}$ inch centre to centre. It is the practice of some of the most experienced makers to allow a V-shaped space under the exhaust steam inlet, as shown at *N*, Fig. 212, so as to allow the steam to distribute itself easily along the whole length of the condensing chamber; otherwise there is a tendency for the cooling effect to be more or less confined to the middle part of the condenser, between the inlet *J* and the outlet *K*.

Tubes. The tubes are made of brass, and, according to the Admiralty requirements, should be 70 per cent. copper, 29 per cent. zinc, and 1 per cent. tin, and a thickness of 17 to 18 Birmingham wire gauge. When used with salt water, or water containing chemical impurities, the tubes are better tinned.

A space will be noticed on the centre line of Fig. 212, in which there are no tubes, the space is to receive the mid-feather *E*, shown in Fig. 213, and it serves to divide the tubes into two groups; the condensing or circulating water enters at *H* into the chamber *D*, passes through the lower nest of tubes into chamber *F*, then by the upper nest to chamber *G*, then out by the exit *N*. By this method the water is compelled to circulate at double the speed that it would have if it went through all the tubes at once from one end to the other only, and, other things being equal, the more rapidly the circulating water moves, the more effective is its action. *L* and *M* are cast-iron covers bolted on to the ends so that they can be easily removed for examining and cleaning the tubes.

Direction of Steam. At one time it was a question whether it was better to take the steam through, or around the tubes, but now the almost universal practice is to take the circulating water through the tubes as shown, and the steam on the outside. As we stated above, the faster the circulating water moves, the better is the vacuum, and, of course, the more water is used, unless the condenser tubes be split up into three or more groups. The only drawback to doing this is the extra power required to draw and force the circulating water. About one foot per second, or, say 60 to 75 feet per minute, is the general practice. The author prefers the lower speed; a better vacuum is got with the latter, but at extra cost, and, excepting for turbines,

a very high vacuum is not to be desired. The greater cooling of the low pressure cylinder and consequent greater range of temperature within the cylinders with reciprocating engines, results in a loss of economy, more than balancing the gain in power. No hard and fast rule can be laid down as to which is the most economical vacuum to work with, but it lies between 25 and 28 inches of mercury, and, in the author's opinion, is nearer the former than the latter.

High Vacuum.

Independent surface condensers are often made connected with their own air and circulating pumps, and this is a correct arrangement when one circulating plant serves for a number of engines. Sometimes it is convenient for one engine. The more economical and general plan is to combine the air and circulating pumps, and work them from the tail rod of the engine, as shown in outline in Fig. 214, where A is the air pump and B the circulating pump. The net power required to work the pump in this way is, of course, less than when an independent engine is used, or when they are driven by an electric motor.

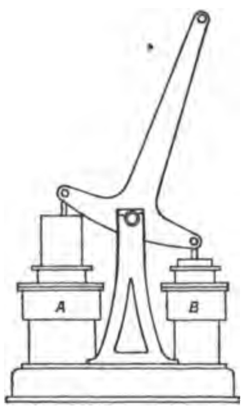


Fig. 214.

There are no generally recognized standard proportions for surface condensers and their accessories, and the practice varies very much, but the following rules may be depended upon to give satisfactory results :—

The size of the tubes should be from $\frac{5}{8}$ to $\frac{3}{4}$ of an inch diameter.

The condensing surface, i.e. outside the tubes, is often given as one square foot per indicated horse power, but as the steam per horse power varies, a better rule is to allow one square foot for each 10 lb. of feed water used.

The quantity of circulating water should be on the average 100 lb. to each lb. of steam to be condensed ; when the water is very cold half this amount can be made to serve, and when warm as much as 125 lb. will be required.

The velocity of water through the tube should be from 50 to 75 feet per minute.

The air pumps should have a capacity of ·6 to ·75 cubic feet per lb. of steam to be condensed. The latter is only required for a very high vacuum.

There are two methods of fixing the tubes in the tube plates, both shown in Fig. 215, where A shows the older method; for this method the holes are bored out about $\frac{1}{4}$ of an inch larger than the outside diameter of the tube $A\,A$, and a slightly tapered tube of wood is driven in moderately tightly. As the wood expands when it becomes wet, it swells out at each end, as shown at $C\,C$, and the tendency to swell in the tube plate makes a good tight joint. This method is quite satisfactory when the condenser is being continually used, and thus the ferrules remain always wet, but when the use is intermittent there is often trouble from leakages after the plant has been standing for some time.

Tube Packing.

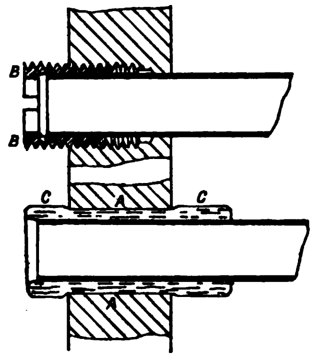


FIG. 215.

The method more generally adopted now is to fix the tubes by means of a screwed gland $B\,B$, the tube plate being bored out first to diameter of tube, and then enlarged and tapped as shown, to form a stuffing-box, a special tape packing being generally used. This makes a very good job and the packing keeps good when dry. There is another advantage this gland has, the small shoulder shown on the inside of the gland B keeps the tubes in position and prevents them moving endwise, as they sometimes do with the wood packing.[1]

Air Condensers. There are many circumstances under which it is advisable and almost essential to condense the steam used by an engine. But sometimes it is impossible to get water either in quality or quantity sufficient either for jet or surface condensation, excepting at a ruinously extravagant cost. A typical case of this kind came under the author's notice some years ago, where a steam engine was working the plant of a copper mine in the desert of Atacama in South America.

[1] Papers giving much valuable information on surface condensers have been written by Professor John Reynolds, *Philosophical Transactions of Royal Society*, vol. cxc; by Dr. Stanton, *Minutes of Proceedings, Civil Engineers*, vol. cxxxvi, part ii; and by R. W. Allan, *Proceedings of the Institute Civil Engineers*, vol. clxi, 1905, part iii.

Rain was almost unknown and the only water obtainable was that obtained from sea-water which had been brought a long railway journey from the coast, but water was indispensable. The engineer in charge had made a rough but effective and exceedingly valuable surface condenser, by connecting together by pipes and tin tubes, a great number of empty tin drums of all sorts and sizes, and disposing them on a slight incline so that they all drained back to the hot well near the boiler and engine house. Thus, though it practically never rained, and during the day there was a temperature in the air of over 120°, leaving only a difference of 90° between the temperature of the exhaust steam and the outer air, yet a very large proportion of the steam was condensed, i.e. about 75 per cent.; this was helped materially by the night temperature, which was often quite cold with the wind from the eternal snow of the Andes, though far away. This was the case of an air condenser pure and simple, and, granted sufficient surface, all the steam used can be recovered in the form of water, less that lost by leakage. At the far end of this condenser was a pipe with an open end to avoid the risk of back pressure, but it was seldom that any steam was seen to issue from it during the day, and never at night. With such a condenser it is, of course, hopeless to expect a vacuum.

The effectiveness of an air surface condenser is immensely enhanced by making and keeping the surface wet. This promotes evaporation from the surface, such evaporation resulting in a lowering of the temperature and causing a more rapid condensation of the steam inside. Such condensers are now made largely and are called evaporative condensers.

Evaporative Condensers. Wherever water is scarce or when it has to be purchased, as is the case in many towns and cities, these condensers are very valuable. The water which has to be used on the outer surface and which is evaporated, is less in amount than the steam condensed into water within them. Thus, though some water is lost by the condenser, the total amount used and paid for, is smaller than if the engine were worked non-condensing, and there is besides, the gain in power and economy from the vacuum.

With this class of condenser and a good air pump nearly as good a vacuum can be obtained as with a water cooled surface condenser. Its principal drawback is the space required for it,

which is much greater than for an injection or surface condenser of equal capacity. This drawback is counterbalanced by the fact that it does not matter much where the evaporative condenser is fixed, the roof of a factory or mill being often utilized for the purpose, thus the expense of additional ground space is avoided.

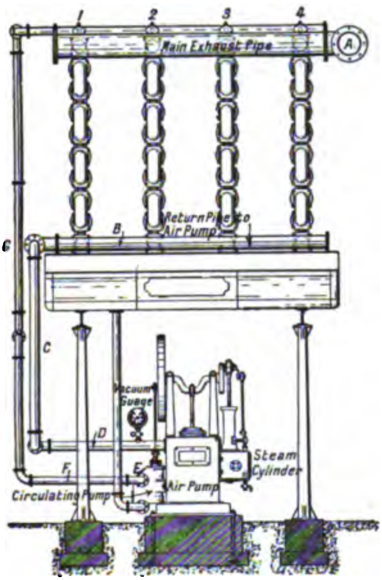


FIG. 216.

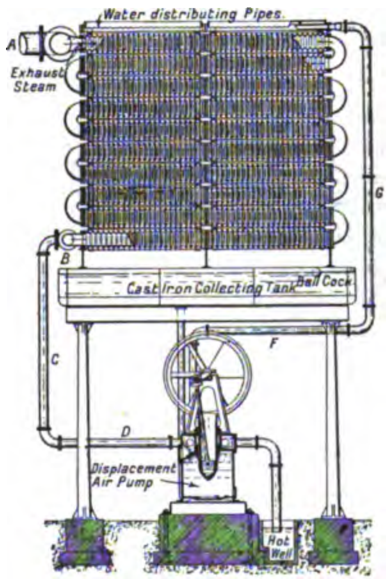


FIG. 217.

This condenser as usually made,[1] consists of a number of gilled pipes made into straight coils and placed to the number of ten, one over the other, as shown in Figs. 216 and 217. The exhaust steam enters by the pipe A through a large pipe direct from the engine. This pipe has four branches corresponding with the four groups of pipes, 1, 2, 3, 4 in Fig. 216 ; each branch connects to a complete coil, as shown partly in section in Fig. 217, and the whole of the ends of the coil again connects to the pipe B and then to the air pump by the pipes C and D. The water for condensing is pumped up by the circulating pump shown at E, and goes by the pipes F and C to the top of the group of pipes, distributing water in a fine shower upon each group ; the water spreads over the gills, which much

[1] Ledward's patent.

increase the evaporative surface of the pipes. A cloud of vapour arises constantly from the pipes, and this water is lost; that which is not evaporated falls into the tank below and is used over again. The plant shown has a total length of pipe of about 500 feet, and can condense 2,500 lb. of water per hour, and would serve for an engine indicating 200 horse power, being 12·5 lb. of steam per H.P. per hour.

This plant is shown self-contained, but very frequently, and quite conveniently, the condenser is on the roof, and the air and circulating pump connected to the engine and worked as shown in Fig. 214.

Considering that with a plant of this kind, less, rather than more, water is required than is needed for a non-condensing engine, it has very much to recommend its use where economy in the use of steam is an object.

Water Cooling. When condensing water is scarce it is quite a common practice to have a large open shallow tank or pond into which the condensing water is delivered as it leaves the engine, and in which it may get cool ready for use again. If merely the surface of the water is exposed to the cooling action of the air, the tank has to be of very large size, so large that space often cannot be spared for it. Another device adopted is to erect high towers with open sides and filled with intertwined wood strips or other material, so as to expose a very large surface and to allow the air free passage through. The water has to be pumped to the top, then trickles slowly down, and gets cool by the time it reaches the bottom. This method is effective, and the tower occupies much less ground space than the tanks. Another and, in the author's opinion, a better way, is to use a small tank and throw the water into the air in the form of a fine spray; this exposes the maximum of surface to the air, and in rising through it and falling down again the water becomes cool.

Ejector Condenser. Another method of producing a vacuum must be noticed, viz. the ejector condenser. Such a condenser requires a large quantity of condensing water, not less than 25 times the amount of the feed water; so that in this respect it is less favourable than the evaporative and but little less than the jet condenser, which requires 28 to 30 times the feed, unless the conical injection described on p. 226, is used, when the amount is equal to about 25 times. The

ejector condenser has one great advantage over others. It has no moving parts, and therefore requires no power to work it; it is also very convenient for use in connexion with quick revolution engines, from which direct-acting air pumps cannot be worked.

For the most successful use of the ejector condenser there should be an ample supply of water with a fall of 28 to 30 feet from the engine.

The construction and working of the ejector can be readily seen from an inspection of the illustration, Fig. 218. Here *A* is the inlet for the exhaust steam, which enters and fills the annular chamber *B B*. From this chamber the steam passes through the many openings into the conical chambers within it, meeting the condensing water at the centre, which enters by the opening *C*. After combination with the steam the water is discharged at *D*. The discharge end of the condenser is connected by a pipe of 15 to 30 feet in length, according to the fall, the end of which pipe is sealed by immersion in a small tank of water to prevent any entrance of air. The speed at which the water falls through the pipe causes an induced current which carries away the air liberated from the steam, thus dispensing with an air pump, and a vacuum of about 25 inches is obtained with a fall of 25 feet.

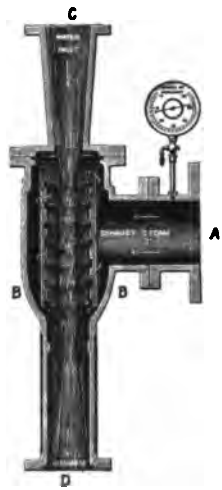


FIG. 218.

This condenser will work efficiently with a difference of height of 15 to 20 feet, the smaller height giving, of course, the lesser vacuum. When there is not a fall away from the engine level, the same result is obtained if there be an equal fall to it from a height above. When there is a supply of water and no fall either to or from the engine, the water may be raised and delivered into the condenser by a centrifugal pump giving a pressure equal to 20 feet head at a cost for power not exceeding 5 per cent. of the gain.

Only a study of each separate locality and a careful investigation into all the circumstances, can enable any one to decide intelligently which method of condensing or cooling is the best, or whether it is worth while to use any of them.

CHAPTER XIII

The Steam Turbine[1]

AMONGST the many forms of the steam engine which have been designed and introduced in modern times none has attracted more attention or has been more worthy of it than the steam turbine. While in its simplest form it dates back to 130 years B.C., yet it is only during the last twenty-five years that it has attained sufficient perfection to be regularly used as a source of power.

Earliest Form. The first engine described by Hero of Alexandria, as is well known, consisted of a hollow globe (Fig. 219) with two bent pipes *a a* issuing from its circumference. When steam was introduced into the globe through one end of its hollow axis, the reaction due to the steam from its nozzles turned the globe round in the direction of the arrows, and a pulley placed on the end of its axle would have been able to give motion to another machine. There is no record of the engine having been so used, though about the year 1577 a German engineer is said to have used one.

The globe in Hero's engine is of no service save to transmit steam to the nozzles; this can quite as well be done by a tube, and if a tube be used bent in the form of S, as in Fig. 220, and enclosed in a casing or cover, we have the essential feature of the reaction turbine as first used by Gustave de Laval about the year 1882. These earliest turbines of Laval were used for driving cream separators, for which purpose their high speed made them very efficient, but they were exceedingly

[1] For much of the information in this chapter, and for many of the illustrations, the author is indebted to Messrs. Greenwood & Batley, the manufacturers of the "Laval," and to Messrs. Parsons & Co., the makers of the "Parsons" turbine, who have kindly placed matter and drawings at his disposal.

wasteful in fuel, though this is not of great consequence considering the very small power required.

Another form of turbine dating from remote antiquity (Fig. 221) consisted of a wheel with vanes upon which a jet

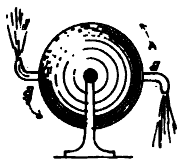


Fig. 219.

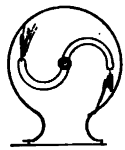


Fig. 220.

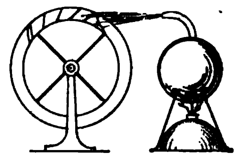


Fig. 221.

of steam played, driving it round. This is the elementary form of Parsons and other "impulse" turbines.

There is a fundamental difference between the ordinary steam engine, in which the steam, working in a closed cylinder, exerts a force in the nature of a pressure, and any kind of turbine, where the steam is used in the form of a strong current of great velocity.

Velocity of Steam. While the molecules of high-pressure steam of, say, 200 lb. above atmosphere, have among themselves an almost infinite velocity, yet the current of steam used behind a piston only moves at the rate of the piston, or from 250 to 1,000 feet per minute. When, however, steam of 200 lb. rushes through a nozzle into the atmosphere, the speed of the current is no less than 186,000 feet per minute.[1]

In many respects the turbine is in the nature of a windmill with many vanes or arms, with the difference, however, that wind, even in the state of the most violent hurricane, does not rush at the rate of more than 10,000 feet a minute (over 100 miles a hour), while steam of 280 lb. pressure rushing into a vacuum goes at the rate of nearly 260,000 feet a minute, and, could the speed be maintained, would rush round the globe in about 8½ hours. It is these tremendous velocities which have to be dealt with in the steam turbine.

Watt's Patent. There have been many attempts to deal with this force, and many patents have been taken out for the purpose, amongst the earliest of them being that of James Watt in the year 1784. From that date up to the year 1884

[1] Parsons estimates the velocity to be about 16 per cent. more than this.

no fewer than 150 British patents were taken out for different forms of turbines, but none of them of great practical value. It was in the latter year that the engineering world was startled by the announcement that a steam engine of 10 H.P. had been made with a working speed of 18,000 revolutions per minute. This was an epoch-making event, and since that date progress has been extremely rapid.

Parsons' Patent. This turbine, patented by the Hon. C. A. Parsons on April 23, 1884, consisted of two groups of fifteen successive turbine wheels or rows of blades 3 inches in diameter on one shaft within a concentric case. On the right and left of the steam inlet the moving blades or vanes were placed in circumferential rows projecting outwards from the shaft, and nearly touching the case; these revolved with the shaft and are called the moving blades; an equal number of fixed or guide blades similarly formed, projecting inwardly from the case and nearly touching the shaft. The surface speed of the turbine blades was about 14,000 feet a minute. This turbine was combined with a dynamo, and both are now in the South Kensington Museum.

General Theory. The general theory is given by Mr. Parsons[1] in his paper as follows :—

"The forces which cause rotation in all steam turbines are the result of the differences of pressure on the internal surfaces, due to the mass and velocity of the steam in its passage through the engine, and not, as in the reciprocating engine, to any statical pressure on moving pistons or abutments.

"The principles governing the efficiency of steam turbines are similar to, but more complex than, those operating in water-turbines.

"The three principal points of difference are :—

"First, in the steam turbine the working fluid is of low density, which implies a high velocity, and necessitates the adoption of high surface speeds of blades if only a few turbines are placed in series on the shaft. Where moderate surface speeds are essential, many turbines must be placed in series.

"Secondly, the working fluid is highly elastic, and obeys the laws of adiabatic expansion of steam when giving out a

[1] *Minutes of Proceedings of the Institute of Civil Engineers*, vol. clxiii, Session 1905–6, part i.

large part of its energy in external work. This renders the use of diverging conical jets essential if the expansion is to be completed in one or in a few turbines. When there are many turbines in series they must be a series of increasing capacity, to allow for expansion.

"Thirdly, although the working fluid may be dry or superheated—and therefore a homogeneous gas—on entering the turbine, yet on account of expansion and the performance of external work a portion of the fluid is condensed into minute drops of water, which are distributed among the remaining gaseous steam; and the working fluid thus becomes heterogenous during the greater part of its passage through the turbine. This admixture of water with the gaseous steam has the effect not only of increasing the surface friction and the resistance in the steam passages, but also, when very high revolutions of the working fluid are adopted, of cutting away the leading edges of the blades—an effect which becomes more pronounced the fewer turbines there are in series."

Modern Example. To follow the development of the Parsons steam turbine from the first of 10 H.P. to the most recent of 70,000 H.P.would take up more space than can be devoted to the subject in this chapter; leaving then the intermediate steps, we will consider the construction of a modern example, a section of which is given in Fig. 222.

Parallel Flow. This is what is known as a parallel flow compound turbine, "parallel" because the steam from its entrance to its exit, flows in a direction parallel to the axis upon which it revolves; and "compound" because starting with a small diameter and fine blades, it, by a series of steps, increases its diameter and enlarges the passages, so as to allow for the expansion and higher speed of the steam in its passage through the turbine.

The steam enters by the passages at *A* and passes to the right through the three groups of blades, each group containing fourteen rings, seven of which are fixed to the casing and are therefore stationary, and seven fixed to the revolving axle. As the steam arrives at the end of each group, to use the words of the inventor,[1] "After leaving the blades it traverses the intervening space in the form of an annular cylinder with a spiral motion, the angle of pitch being about 30° to a plane

[1] Lecture before the Royal Institute of Great Britain, May, 1906.

[Plate II.

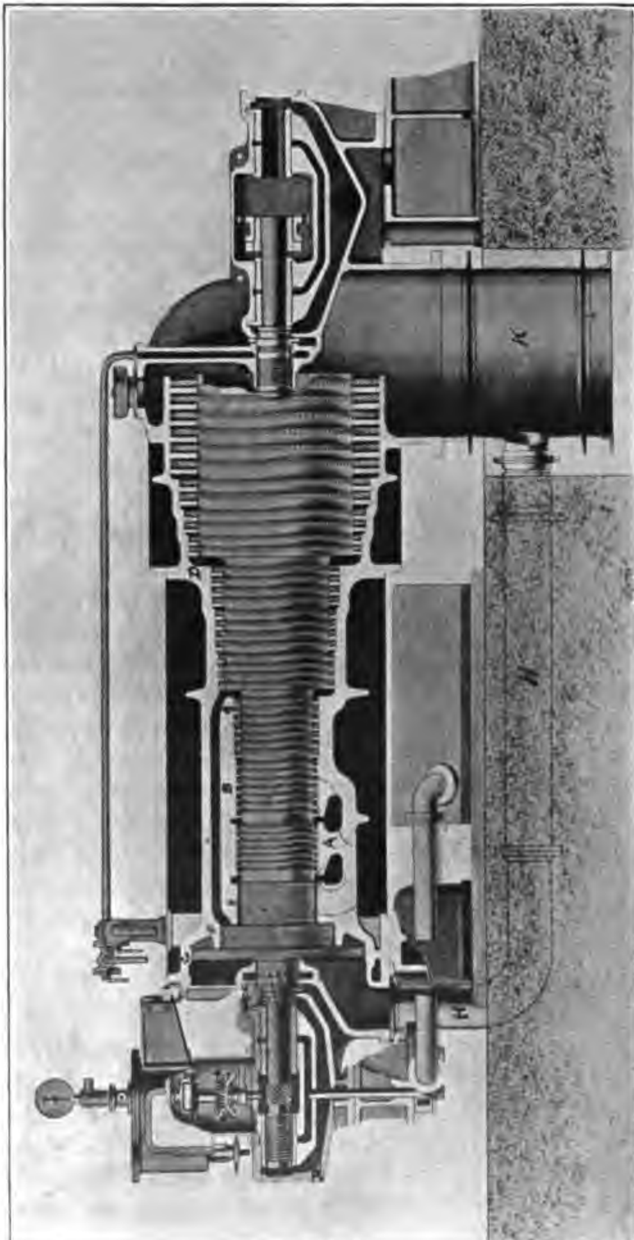


Fig. 222.

normal to the axis; and, as the succeeding blades are moving in a similar direction to this flow, we see that the velocity with which the steam is cut by their frontal edges is much less—in fact, less than one half the velocity at which the steam has issued from the previous blades. From this we see how small is the loss due to the cutting of the steam by the frontal edges in the compound turbine, and also how small is the velocity with which drops of water strike the metal of the blades. This is an important feature.

Erosive Power of Water. "It has been shown by experiments that if drops of pure water, arising from the condensation of expanding steam, impinge on brass at a greater velocity than about 500 feet per second, there results a slow wearing away of the metal. It is very slow, and would require about ten years to erode the surface to a depth of $\frac{1}{32}$ of an inch. In the compound turbine, the striking velocity is much below this figure, and the preservation of their form and smoothness of surface has been found to be practically indefinite.

Erosion of Blades. "It appears that the erosive power of drops of pure water moving at high velocity increases rapidly with the velocity; it may probably be as the square. Experiment has shown that if saturated steam at 100 lb. pressure be allowed to flow through a divergent jet into a good vacuum, attaining a velocity of about 4,500 feet[1] per second, and allowed to impinge on a stationary brass blade, the blade will be cut through in a few hours, and the hardest steel will be slowly eroded. The action seems to be the result of the intense local pressure from the bombardment of the drops, which may exceed 100 tons.

"Owing to the receding velocity of the blades from the blast, and consequently reduced striking velocity, the erosion of the blades in impact turbines is much reduced, and in compound turbines there is complete immunity from such erosion."

Size and Disposition of Blades. "The comparative size and disposition of the blades is shown in Fig. 223, where one set of blades are fixed to the casing and the other set revolves with the shaft. As the diameter increases, the blades increase in size and length and have wider spaces between them to allow

[1] Other authorities give the velocity at 3,871 feet per second; both of them can only be approximate. (Author's note.)

of the full expansion of the steam. At the exhaust end the blades are made much larger and are connected together at the outer ends by a ring, so that they are more like a set of louvres than individual blades. A small part of one of these large rings is shown in Fig. 224. About 50,000 blades are used in such a turbine as shown. On examining Fig. 222, it will be

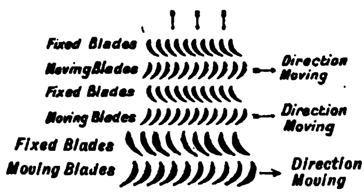


FIG. 223.[1]

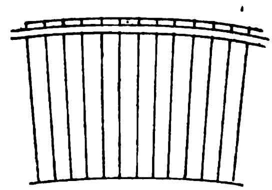


FIG. 224.[1]

seen that commencing at *C* there is a long port leading to the left. This port conveys high pressure steam which has passed the first two sets of turbines to the guiding blades or rings of the turbine *E*, *F*, *G*. Those at *E* are exposed to the full pressure of the entering steam, but are so well fitted and so numerous that there is no leaking past them. The steam entering by the port to the face of the next set of rings, *F*, is of lower pressure, as it has undergone some expansion; it presses against the surface of *F*, and any escape past the grooves fills the next space, and, with still lower pressure, presses against the large surface at *G*. Any leakage past *G* enters the chamber shown to the left of *G*, and then by the pipe *H H* to the large exhaust pipe *K*, from which it passes to the condenser. These guiding blades and grooves serve to keep the turbine shaft in its exact position longitudinally, and by their surfaces, exposed on the one side to the steam pressure and on the other to the vacuum, they tend to balance the longitudinal pressure, and thus materially reduce the internal friction.

Balancing.

Leakage. There are some losses in the steam turbine, especially that of leakage of steam past the blades, which are greater than in connexion with reciprocating steam engines. Yet notwithstanding this fact, when of large size

[1] Each blade is of very small size. The full size of them can be got by multiplying the illustration by 2·5.

they use no more steam than do the latter when they are of equal power and working under equal conditions. The principal cause of this, in the author's opinion, is because they are better fitted to deal with great extremes of temperature. We have shown in chapter IV how great is the loss from the fall in temperature, and how it can best be dealt with by dividing the expansion between two or more cylinders, and avoiding too great a vacuum in the condenser, because the fall in temperature produces losses greater than could be compensated by the gain in power. It is, however, a fact well established by careful experiment that the greater the vacuum with a turbine, the more economical is the engine: this is because with a turbine there is no alternation of heat and cold. The turbine may start with steam of any pressure and temperature, and the admission end will always stay at or near the temperature of the admission steam. Further, the exhaust end will remain always at or near the temperature of the vacuum, hence the great source of loss in the reciprocating engine is avoided. There is, of course, a steady and continuous transmission of heat by conduction from the hot to the cool end of the turbine, but this is very small compared with the loss from alternate heating and cooling.

Small Temperature Losses.

Lubrication. The bearings of the turbine are the only parts in which there is any mechanical contact; these are kept well supplied with oil from a pump which maintains a constant pressure of from 3 to 5 lb. per square inch all round the shaft, and so long as this is maintained there can be no real metallic contact, but only a thin film of oil between the shaft and its bearings which enables them to wear for many years without any adjustment. This pump is driven by worm gearing from the shaft so as to reduce its speed.

Governors. The governor is shown in the chamber to the left (Fig. 222); it is a highly sensitive spring governor, and it actuates the steam admission by means of a steam relay, i.e. there is not a constant admission of steam at a varying pressure, but a cam on the oil-pump opens and shuts the admission valve, thus causing the admission of steam to be periodic, in the nature of a succession of gusts, the duration of each gust being determined by the position of the centrifugal governor. By this means whatever steam is admitted is

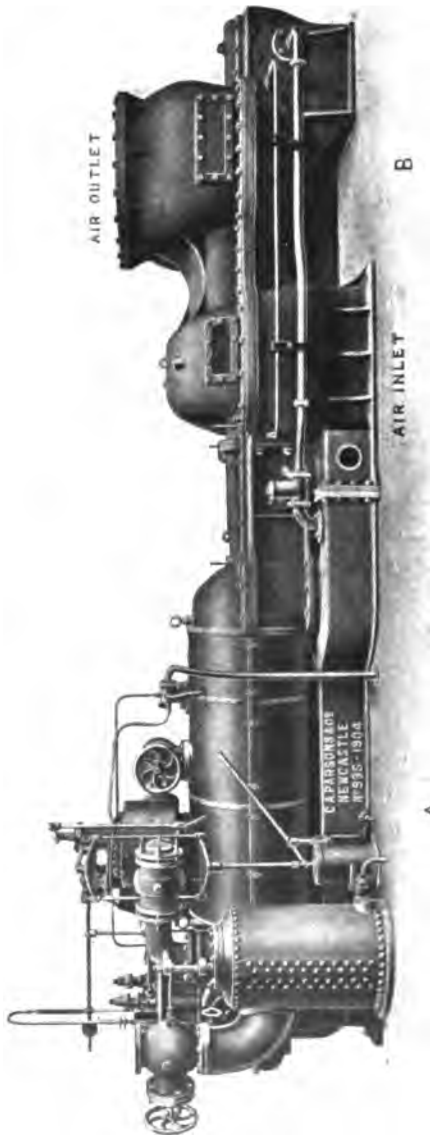


FIG. 225.

always at full boiler pressure, even when the turbine is running at low speeds and with reduced power.

Overload. A bye-pass valve is generally arranged on the steam turbine, by opening which, high-pressure steam is admitted to the low-pressure portions of the turbine, so that its power may be temporally increased to from 40 per cent. to 50 per cent. above normal.

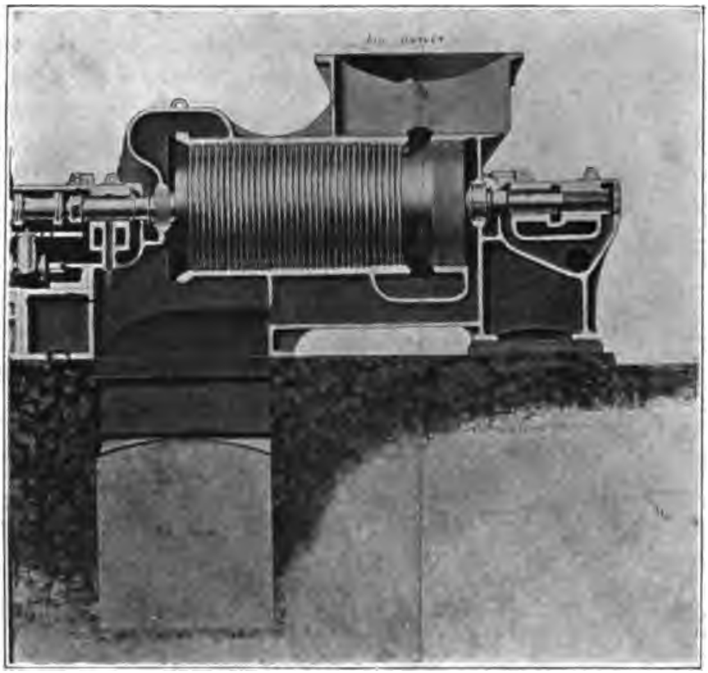


FIG. 226.

Air Compression. An engine which is worked by currents of steam can evidently be utilized to produce currents of air by a reversal of the process. Thus the steam turbine makes a very successful air-compresser or blower; such a one is shown in elevation in Fig. 225. *A* is the steam engine of similar section to Fig. 222, and *B* is the blower, a section of which is shown in Fig. 226. The very great size of the inlet and outlet in the illustration shows the great capacity of this form of blower in proportion to its working cylinder. The cylinder is fitted as seen with fixed and moving blades like

the steam cylinder, but in the case of air, all the rings are of the same diameter, the difference in pressure between the air as it enters at 15 lb. pressure and its discharge at say 25 lb., being only 10 lb. increase. A comparatively small difference in the speed (which can be easily made) within limits of about 15 per cent. up or down, can vary the pressure of the air as much as 50 per cent., or, keeping its pressure constant, can vary the volume delivered.

The small space occupied by a turbine blower engine is seen in Fig. 225. The weight of a complete turbo-blowing engine capable of delivering 23,000 cubic feet of air per minute at 10 to 15 lb. pressure is only about 25 tons. A complete installation of such a blower with its condenser and air pump is shown in Fig. 226A.

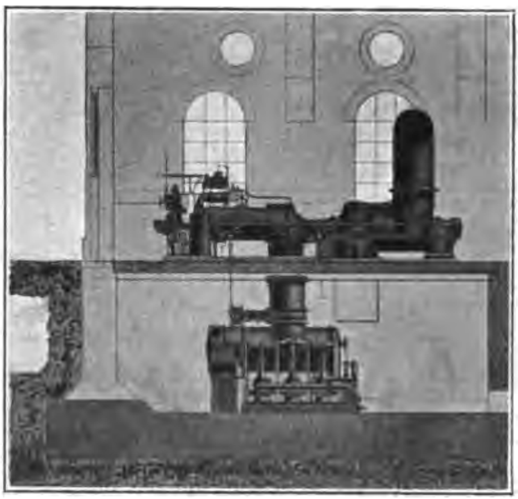


FIG. 226A.

Electric Generator. The earliest purpose to which steam turbines were applied was the generation of electricity. Up to the time of its production there had been a difficulty in getting direct coupled reciprocating engines to revolve continuously at a sufficiently high speed to get the full output from a generator of given size, in other words the generator had to be made large to suit the speeds of the engine. An opposite difficulty presents itself with the turbine. There

is no practical limit to the speed of the engine, but it is extremely difficult, if not impossible, to make a dynamo, the drum of which can be made to revolve safely at a surface velocity of 500 feet per second. Hence either the turbine must be geared down to the dynamo, or made of a much larger diameter than is necessary for a given power, and thus be made both heavier and more costly. In the author's opinion for powers under 1,000 H.P., there are many forms of steam engines which can successfully compete with the steam-turbine. When dealing only with cost, it must be remembered that, other things being equal, the higher the speed of an electric generator for a given output, the lower will be its cost, and the greater the diameter and lower the speed of a turbine, the higher will be its cost. Notwithstanding this, turbo-generators have been largely used for generating electricity during the last twenty years, and its makers claim that, for a H.P. of 1,000 to 12,000 and upwards, the steam turbine with its accompanying dynamo, is found to be cheaper in first cost, running expenses and fuel, than the reciprocating engine and its slow-speed dynamo. This is so far true that it seems possible to generate electricity in colliery districts almost, if not quite, as cheaply for electro-chemical purposes, as it can be produced at Niagara and some other large centres of water power.

As an example of the economy of a Parsons' turbine, we give particulars of a test of a 3,500 K.W. Turbo-Alternator taken by Mr. C. H. Mertz at the Carville power station, Wallsend-on-Tyne, February 17 of this year (1907). The test was carefully made at no load up to 6,921·8 K.W., the speed being given as constant for all loads at 1,200 revolutions per minute. It is doubtful if much better result could be obtained than from this plant, as the power was great, the steam pressure high, 199·9 practically 200 lb. there were 120·5 degrees of superheat in the steam, and a vacuum of 29·039 inches. Eminently, therefore, no fear of high economy, which was certainly reached, the best result being got from the No. 7 experiment with a load of 5,164 K.W. and a steam consumption (as measured by the works condenser) of 13·189 lb. per K.W. per hour, corresponding to 9·839 lb. per H.P. per hour.

Test No.	Duration (Hrs.)	Mean Calibrated K.W.	Steam. Pressure.	Steam. Temperature at Turbo. °F.	Steam. Superheat °F.	Speed R.P.M.	At Turbo Exhaust. Vacuum at 30 in. Brm.	Water. Total Condensed Lbs.	Water. Lbs. per Hour.	Water. Lbs. K.W. Hour.
1	½	No load not excited.	180	460	80	1,200	28·875	1,835	3,670	—
2	½	No load excited.	211	453·3	61	„	28·95	2,603	5,206	—
3	1	2,192·87	202·4	492·1	103	„	29·036	31,836	31,836	14·517
4	1½	4,045·14	197·4	495	108	„	29·066	83,972	55,981·3	13·839
5	1½	5,901·	195·8	503·2	117	„	28·95	119,182	79,454·6	13·464
6	½	6,921·8	198·4	505·5	118·5	„	28·765	47,390	94,780·	13·692
7	1	5,164·07	199·9	508·5	120·5	„	29·039	68,180	68,180·	13·189
8	3	5,059·38	194·5	477·9	92	„	29·195	203,559	67,853·	13·411

Ship Propulsion. The most striking success of the Parsons' turbine is its application to ship propulsion, for which it has many decided advantages. Prominent amongst these are, the smaller space required, the lesser weight to be carried, and the very great reduction in vibration. It was as recently as 1897 that the series of experiments were made with the **The "Turbinia."** *Turbinia*, which established the success of this form of steamship propulsion.[1] Since that time progress has been very rapid. This can be best seen by comparing

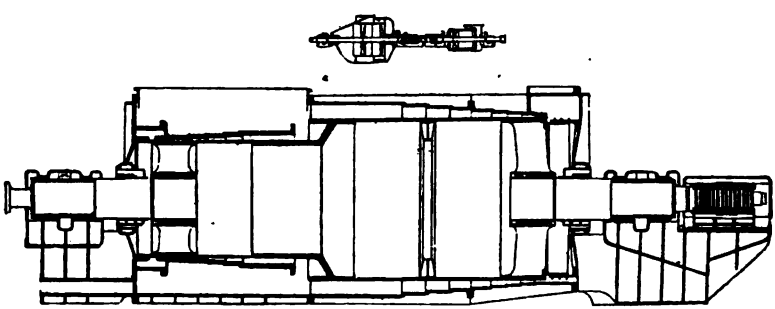


Fig. 227.

the outline drawing of the *Turbinia* with the outline of the engines of the Atlantic liners *Lusitania* and *Mauritania*

[1] A full account of these trials can be found in the *Transactions of the Institution of Naval Architects*, vol. xlv (1903), p. 284.

launched this year, 1907 (Fig. 227), or, better still, a diagram to scale of the ships themselves which are shown in Fig. 228.

The arrangement of the turbine in the vessels is clearly shown in the diagrams, Fig. 229, showing the end elevation,

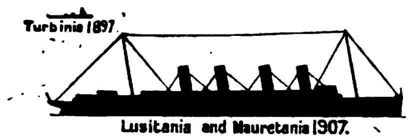
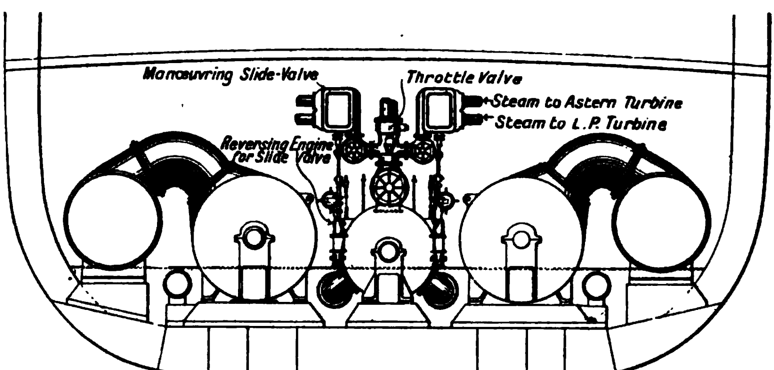


Fig. 228.

and Fig. 230 showing the plan of the engine room of one of these Atlantic liners. These diagrams are prepared from drawings used to illustrate Mr. Parsons' paper on the steam turbine read before the Institute of Civil Engineers in 1905[1] and kindly furnished to the author for this work. It will be seen that for propelling these ships there are five turbines used, connected to three screw propellers. The high-pressure turbine is coupled direct to the centre screw for forward motion only. The high-pressure exhausts into the two low-pressure turbines, one on either side, each coupled direct to a screw propeller. On the same shaft as these latter is another stern



Cross Sectional Elevation.

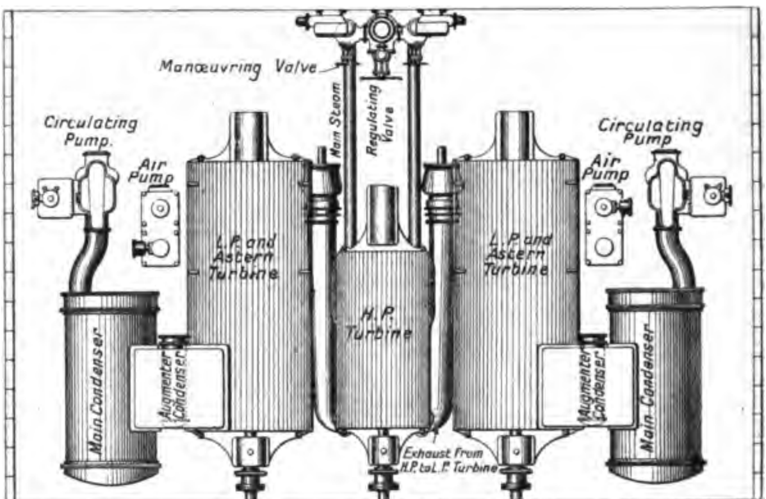
Fig. 229.

turbine. When the vessel is going forward, which of course is for the greater part of its time, all these turbines are driving

[1] *Minutes of Proceedings of the Institution of Civil Engineers*, December 5, 1905.

forward, the two astern ones running idly; there is no loss by mechanical friction from this cause, as all parts revolve free from contact. When going astern the two low-pressure turbines only work, steam being admitted to them by a bye-pass valve. For manœuvring, one of the low-pressure turbines can be run in one direction and the other in the contrary. All the parts are so clearly shown in the diagram that further description is not needed.

The using of the steam so expansively by the enlargement of the parts of the turbine as the steam passes through, is



Plan.

FIG. 230.

doubtless one of the causes of the turbine's success and economy, and too much credit cannot be given to Mr. Parsons for the sk'll and ingenuity with which he has made his designs and perfected the details of his invention.

In the twenty years following the public demonstration of the success of the first turbine in 1884 nearly a hundred patents for turbines were taken out in England alone without counting the Continent and America. Amongst the number was one by the author in 1887,[1] from the provisional specification of which the following extract is made :—

[1] Patent No. 10,005 (Richardson), July 16, 1887.

"This invention related to a novel method of obtaining rotary motion by the tangential force of issuing steam, also to a method of utilizing the exhaust from one disc or cylinder to another, to produce motion in one of greater capacity surrounding the first.

Richardson's Patent.

"For the purpose of my invention :—The steam is introduced into the centre of the disc or cylinder mounted on a spindle, round the circumference of which are a number of openings arranged as nearly as possible at a tangent to its circumference and communicating with the central chamber; the steam leaving these tangential orifices at a high velocity, by its reaction drives the disc or cylinder backward and thus gives rotary motion to the spindle upon which it is fixed. On leaving the disc, the steam instead of being discharged into the atmosphere enters the interior of another disc surrounding the first one, from whence it emerges also in a tangential direction, thus giving rotary motion to the second disc, which is mounted upon a sleeve on the spindle.

"The second disc being larger in diameter and capacity, the steam leaves it at a lower pressure than the first; this can again be surrounded by a third or any number until the escaping steam is reduced to atmospheric pressure, every alternate disc being attached to the spindle and the other to the sleeve. Gearing may be attached to connect the two together, or if required one may be fixed, in which case the other would revolve at double speed."

In making the experimental engine one set of blades was attached to the revolving disc and the other set was fixed to the casing; owing, however, to the low pressure of steam used and the large size of the blades and openings no satisfactory results were obtained, and the inventor being then very fully occupied with other engineering matters let the disc turbine drop, and the patent was not completed.

This form of construction, however, possesses many advantages; a very high peripheral velocity is obtained with a very small weight, and the material being in the form of a steel disc is of the best form so far as strength of materials is concerned. Such a turbine is also in perfect balance in all its parts, as is seen in Fig. 231, steam being admitted on both sides of the revolving disc; this latter works in equilibrium so far as pressure is concerned. It is not improbable that progress

will be made with this method of construction in the near future. The author has since learned that "Wilson" in

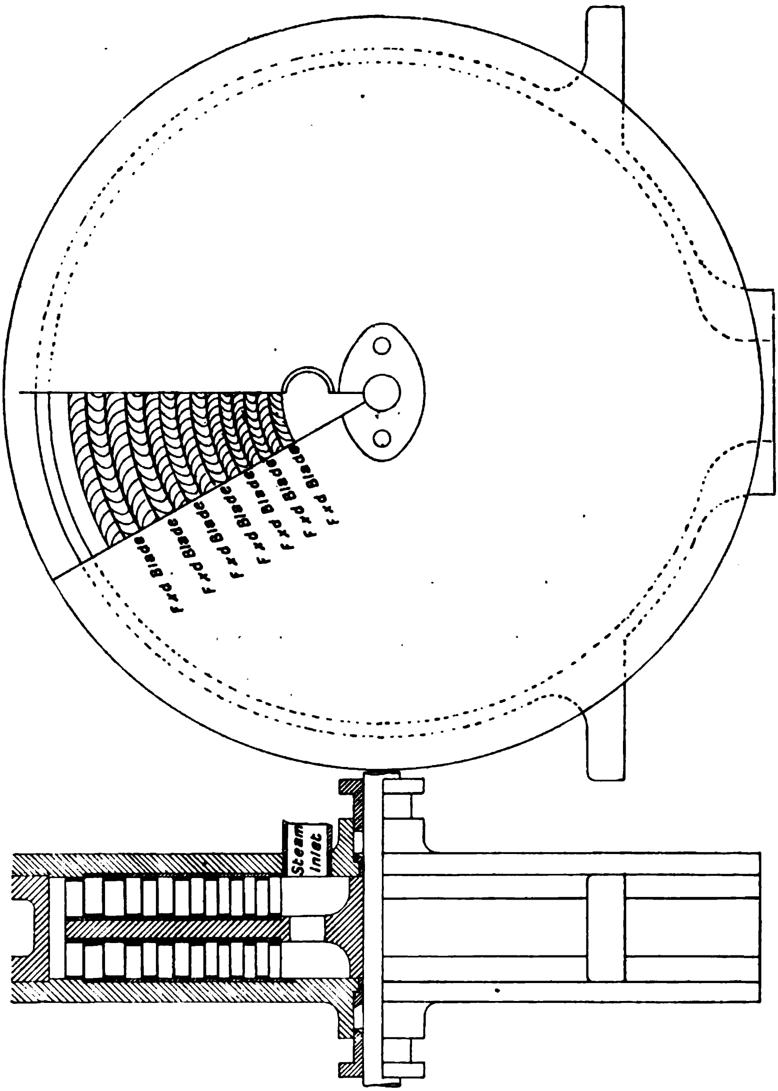


FIG. 231.

1848 patented a turbine having some features in common with that above described. We have no record of any of its

performances, and it was not balanced by having blades on both sides of the disc.

Many of the patents taken out by Parsons having expired by efflux of time, there are now numerous manufacturers, especially on the Continent and in America, many more than we have space to deal with. The one other which in some respects may be considered a rival in success, is that of De Laval, which, in its present general form, was patented in 1886, since when several other patents have been taken out for modifications and improvements.

De Laval Turbine.

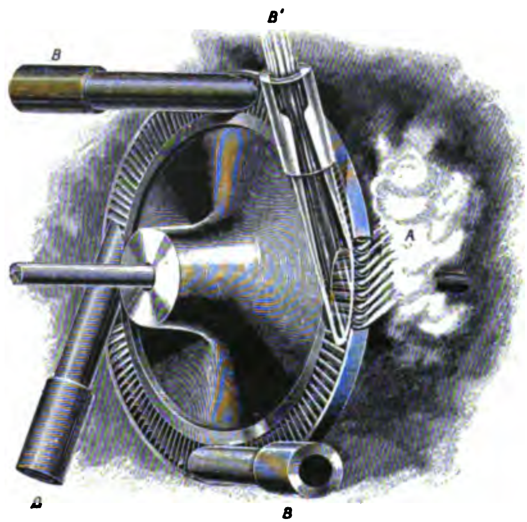


Fig. 232.

It is totally different from the Parsons' or others of that type, and has many advantages distinctively its own. In this turbine the distinctive feature is the "expanding nozzle," a description and diagram of which, with other particulars, has been kindly supplied by the makers for this work.

In the turbine as now constructed, the steam is blown from stationary nozzles against the vanes of a revolving wheel. The steam passes through the vanes or buckets of the wheel, delivering most of its energy to the wheel, and is afterwards exhausted to the atmosphere or to a condenser in the usual

way. The turbine wheel is constructed with one row of buckets only. The energy of the steam is taken up by the turbine wheel and transmuted into rotary motion. The illustration (Fig. 232) give a general idea of the arrangement.

This illustration shows all the essential features of the Laval turbine. The wheel of steel is shown with its one set of curved blades or buckets surrounded by a thin ring or cover which enclosing all the buckets compels the steam entering at one side of the wheel to depart from the other side at *A A*, where the steam is issuing; the cover is removed so that the shape of the blades can be seen better. *B B B* and *B′* are four nozzles which transmit the steam from the supply pipe to the turbine wheel; *B′* is shown semi-transparent. The steam enters through a pipe of large diameter at *B′*, and at

Expanding Nozzle.

full boiler pressure of say 200 lb. A little below its entrance the passage is contracted to a small hole, the steam in rushing through which is reduced in pressure to nearly ½; from this point the hole is enlarged to allow for the expansion of the steam, which falls in pressure rapidly, but at the same time increases proportionately in speed, until it attains enormous velocity, and thus has a high kinetic energy or moving power.

This high speed is obtained by expanding the admission steam in the above nozzles, which are specially constructed for the purpose. Thus if the turbine works with steam of

Velocity.

200 lb. pressure above atmosphere and exhausts into a vacuum of about 28 inches, say 1 lb. pressure absolute, the steam would be expanded from 215 lb. to 1 lb., and would attain a velocity of over 4,000 feet per second.

According to Professor Zeuner, who has made very extensive tests on the outflow of steam, if the steam is expanded adiabatically in the nozzle, all the potential energy of the steam is transformed into kinetic (or moving) energy, and this kinetic energy is absolutely identical with the amount of work which the same steam would have done if it had been expanded in the same proportion in the cylinder of an engine.

The table below gives the velocity of steam at different admission pressures expanded in nozzles down to atmosphere

in Table I, down to 25 inch vacuum in Table II, and down to 28 inch vacuum in Table III.[1]

THE VELOCITY OF OUTFLOW AND THE WORKING CAPACITY OF DRY SATURATED STEAM.

Initial Steam Pressure, lb. per square inch.	I. Counter-pressure 1 atm.			II. Counter-pressure 2·4 lb. per sq. in. absolute corresponding to 25 in. Vacuum.			III. Counter-pressure 0·93 lb. per sq. in. absolute corresponding to 28 in. Vacuum.		
	Velocity of Outflow of Steam, ft. per second.	Kinetic Energy, ft.-lb. per sec.	H.P. of 550 ft.-lb. per. sec.	Velocity of Outflow of Steam, ft. per second.	Kinetic Energy, ft.-lb. per sec.	H.P. of 550 ft.-lb. per. sec.	Velocity of Outflow of Steam, ft. per second.	Kinetic Energy, ft.-lb. per sec.	H.P. of 550 ft.-lb. per sec.
		Per lb. of Steam per hour.			Per lb. of Steam per hour.			Per lb. of Steam per hour.	
60	2,421	25·29	0·046	3,320	47·57	0·087	3,680	58·44	0·106
80	2,595	29·06	0·053	3,423	50·56	0·092	3,793	62·08	0·113
100	2,717	31·86	0·058	3,520	53·47	0·097	3,871	64·66	0·118
120	2,822	34·37	0·062	3,596	55·80	0·101	3,940	66·99	0·122
140	2,913	36·62	0·066	3,661	57·84	0·105	3,999	69·01	0·125
160	2,992	38·63	0·070	3,718	59·65	0·108	4,045	70·61	0·128
180	3,058	40·35	0·073	3,764	61·14	0·111	4,091	72·22	0·131
200	3,115	41·87	0·076	3,810	62·64	0·114	4,127	73·50	0·134
220	3,166	43·26	0·079	3,852	64·03	0·116	4,159	74·64	0·136
280	3,294	46·83	0·085	3,962	67·74	0·123	4,229	77·18	0·140

It will be seen from the above table how exceedingly high is the velocity of the outflow of steam when expanded adiabatically in suitable nozzles. Thus taking the last figures in the table, steam of 280 lb. pressure expanded down to a vacuum of 28 inches leaves the nozzle with a velocity of

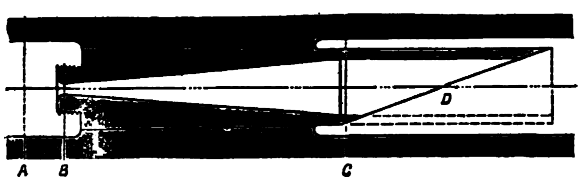


FIG. 233.

4,229 feet per second, a speed at which it could travel round the world in about 8½ hours.

The construction of the nozzle is very simple and can be

[1] According to Parsons the velocities are about 15 per cent. higher than this; thus probably the truth lies between the two estimates.

easily understood by examining Fig. 233, from which it will be seen that the steam enters at *A*, the passage up to this point being about ten times larger in diameter or 100 times greater in area than that at *B*, which is the smallest size of nozzle; from this point to *C* the passage is enlarged, the shape being a perfect cone; from *C* to the end it is a cylinder of a diameter equal to large end of cone. The cylindrical part is then cut at an angle of 20°, shown by the line *D*, and it is this part *D* which is fixed in the casing surrounding the wheel so as to be only 2 mm. from the wheel's face, as shown in Fig. 232. The proportions of the nozzles are as follows: At *A* there must be full boiler pressure, and this part is generally enclosed in a steam chamber of large capacity. The diameter of the small section *B* and the large section *C* are to each other as 5·2187 to 1, and thus the area of *C* will be the square of this or 27·2348 times larger than *B*. If for instance the diameter of the small section *B* is 6 mm. the diameter of the large section will be 6 × 5·2187 or 31·312. Through such a nozzle there passes 479 lb. of dry saturated steam of 200 lb. pressure per hour neither more nor less. This fact of the nozzle passing only a certain amount of steam per hour is often used to ascertain the steam consumption of the turbines. Supposing that the admission steam is dry, i.e. that it does not contain any moisture, then in the different sections of the nozzle with steam of 200 lb. pressure and 28 inch vacuum the pressure and quantity of the steam will be as follows:—

Proportions of Nozzles.

SECTION A:—
Pressure: 200 lb. per square inch above the atmosphere.
Percentage of moisture: 0.
Specific quantity of steam: 1.

SECTION B, the smallest section of the nozzle:—
Pressure: 110 lb. per square inch above the atmosphere.
Percentage of moisture in the steam: 4 per cent.
Specific quantity of steam: 0·96.
Velocity of the steam: 1,500 feet per second.
Specific volume of the steam: 3·5 cubic feet per lb.

SECTION C, the largest section of the nozzle:—
Pressure: (28 inches of vacuum) 2 inches of mercury absolute pressure.

Percentage of moisture in the steam : 24 per cent.
Specific quantity of steam : 0·76.
Velocity of the steam : 4,127 feet per second.
Specific volume of the steam : 256·8 cubic feet per lb.

It is evident that as much as possible of the kinetic energy of the steam-jet issuing from the nozzle should be taken up by the turbine wheel, and thus transformed into mechanical energy. In order to obtain the maximum efficiency, the peripheral speed of the turbine wheel should be 47 per cent. of the velocity of the steam. The absolute velocity of the steam leaving the buckets is then 34 per cent. of the initial velocity and the energy absorbed by the turbine wheel is 88 per cent. of the kinetic energy of the steam.

If, for instance, the speed of the steam entering the buckets is 4,000 feet per second, the speed of the steam leaving the buckets should be 1,360 feet per second, and the kinetic energy of the steam transformed into mechanical energy is theoretically equal to a consumption of 9·1 lb. of steam per H.P. per hour.[1]

Consumption of Steam. This is a low theoretical consumption, but as we showed in chapter IV a lower theoretical consumption than this is got with a triple-expansion reciprocating engine; owing to its fewer parts and absence of friction, it is not improbable that the turbine may give out a higher percentage of actual to theoretical work.

From tests made by Messrs. Dean & Mann, engineers, of Boston, U.S.A., of a 300 B.H.P. Laval turbine in June, 1902, and published by the Laval Company, we extract the following results :—

No. of Nozzles Open.	Loads.	Relative Loads.	Speeds.	Difference in Speeds.
8	352 H.P.	100%	750 r.p.m.	—
7	298 ,,	85%	756 ,,	$+\frac{8}{10}$ of 1%
5	196 ,,	56%	745 ,,	$-\frac{7}{10}$,, 1%
3	119 ,,	34%	751 ,,	$+\frac{1}{10}$,, 1%

[1] See Zeuner's *Theoria der Turbinen* and the formulae de Saint-Venant and Wantzel (1839).

The loads varied considerably, but the speed was maintained with great regularity, only varying six revolutions in 750, as shown above.

No test seems to have been taken of the number of revolutions under no load.

When tested with saturated steam the result was as follows:—

Saturated Steam

Nozzles Open.	Loads B.H.P.	Relative Loads.	Steam per B.H.P.	Increase for Diminishing Loads, referred to Maximum Load.
8	333 H.P.	100%	15·17 lb.	—
7	285 „	86%	15·56 „	2·6%
5	195 „	59%	16·54 „	9·0%
3	119 „	36%	16·40 „	8·1%

With superheated steam the result was better, as shown below:—

Superheated Steam

Nozzles Open.	Loads B.H.P.	Relative Loads.	Steam per B.H.P.	Increase for Diminishing Loads, referred to Maximum Load.
8	352 H.P.	100%	13·94 lb.	—
7	298 „	85%	14·35 „	2·9%
5	196 „	56%	15·53 „	11·4%

In both cases the turbine exhausted into a condenser, which maintained a vacuum of 27·2 inches. It will be noticed that the economy is greatest when the load is highest. These turbines had twelve nozzles but only eight or fewer were used, thus it had a large margin of power. Each nozzle has an economy all its own, but the power naturally varies with the number of nozzles used, while the losses by friction of parts are constant, hence the higher economy at high duties. Very many experiments were made extending over three days; one such at full load is given below:—

TESTS WITH SUPERHEATED STEAM.

Number of nozzles open, eight (8).
Average reading of barometer, 30·18 in.
Average temperature of room, 83° F.

Date. 1902.	Hour.	Weight of Steam used per Hour. Lb.	Pressure above Governor Valve. Lb.	Pressure below Governor Valve. Lb.	Vacuum. In.	Superheat above Governor Valve.	Revs. per Minute of Generators.	B.H.P.	Steam used per B.H.P. per Hour. Lb.
	A.M.								
May 22	8– 9	4,833	208·3	200·6	27·2	81° F.	—	356·6	13·55
,,	9–10	4,936	207·5	199·3	27·2	86° F.	—	355·7	13·88
,,	10–11	5,083	207·7	202·1	27·2	91° F.	—	357·8	14·21
,,	11–12	4,976	208·3	199·4	27·2	88° F.	—	354·1	14·05
	M. P.M.								
,,	12– 1	4,841	207·5	194·3	27·3	82° F.	—	343·5	14·09
,,	1– 2	4,768	206·9	195·6	27·2	75° F.	—	344·4	13·84
Independent Average	8– 2	4,906	207·0	198·5	27·2	84° F.	750	352·0	13·94

These are exceedingly good results, far above those obtained from the average reciprocating steam engine but not yet equal to what can be got from the highest type of reciprocating engine. For driving high-speed machinery it is not improbable that the economy in transmission might compensate for the difference in steam consumption.[1] The speed in revolutions per minute of the turbine must necessarily always be high. With a velocity of the steam jet of 4,000 feet per second, the speed of the turbine wheel ought to be about 1,880 feet per

[1] For duties up to 1,000 H.P. the reciprocating engine is for many purposes more economical than the turbine, though above this power the advantage seems to lie with the latter, and it remains in its best working condition for a longer time than the reciprocating engine as there are no valves or pistons to wear and get out of order. As a combined plant, i.e. motor and electric generators together, it gives in large power better results than the reciprocating engine working with the same steam pressure. Dr. A. Stodol, of Zurich Polytechnical, in his comprehensive and valuable work on steam turbines, gives a series of carefully conducted tests, and points out that the steam turbine working with moderate superheat has overtaken the "Compound" reciprocating engine . . . . . but after further illustration states that "the triple expansion steam engine has therefore still an advantage of about 8 per cent. of the steam consumption" over the turbine.—Edition, 1906, p. 319.

second, but for several practical reasons the speed is much lower. At the present time the peripheral speed of the De Laval turbine does not exceed 1,380 feet per second; this corresponds to a theoretical consumption of 9·8 lb. of steam per H.P. per hour. The following table gives the actual working speed of some of the standard types :—

SPEEDS OF THE TURBINE WHEELS.

Size of Turbine.	Middle Diameter of Wheel.	Revolutions per Minute.	Peripheral Speed, Feet per Second.
5 H.P.	100 mm. 4 in.	30,000	515
15 ,,	150 ,, 6 in.	24,000	617
30 ,,	225 ,, $8\frac{7}{8}$ in.	20,000	774
50 ,,	300 ,, $11\frac{3}{4}$ in.	16,400	846
100 ,,	500 ,, $19\frac{3}{4}$ in.	13,000	1,115
300 ,,	760 ,, 30 in.	10,600	1,378

It will be noticed from the above table that while the revolutions per minute are very high in the small turbines, the peripheral velocity is low and as a consequence the economy is not so high. In this respect a turbine is like a reciprocating engine, the higher the power, other things being equal, the greater the economy. The following table shows the result of a test with a 50 H.P. turbine with steam pressure varying from 113·8 lb. to 74·0 lb.

RESULTS OF TESTS WITH DE LAVAL STEAM TURBINES AT DIFFERENT LOADS.

Turbine Machine.	Pressure of Admission Steam. Lb. per sq. inch.	Vacuum Inches of Mercury.	No. of Nozzles open.	Electrical H.P.	Lb. of Steam per Electrical H.P. per hour.	Remarks.
50 H.P. Turbine	113·8	26·3	6	49·4	24·6	Work for
Dynamo.	113·8	26·3	5	40·2	25·2	Condensing
The test made in	93·9	26·9	4	25·0	27·9	included.
April, 1895.	74·0	27·5	3	12·7	32·5	

These results are given in electrical H.P.; in indicated H.P. they would be better; thus assuming an efficiency of 85 per cent. the consumption of steam would be in the four cases 20·9, 21·3, 23·7 and 27·6 per indicated H.P. per hour.

With a smaller engine of 30 H.P. the following table shows a still higher consumption of steam. In this case there was also a much lower steam pressure and the steam was exhausted into the atmosphere, there being no condenser.

TESTS WITH A 30 H.P. STEAM TURBINE WORKING WITH SATURATED AND SUPERHEATED STEAM RESPECTIVELY.

NON-CONDENSING.

Steam Pressure: 7 atmosphs. absolute = 88·2 lb.
Speed of driving shaft: 2,000 revs. per minute.
Speed of turbine wheel: 20,000 revs. per minute.

		Half Load.		Full Load.	
		Saturated Steam.	Superheated Steam.	Saturated Steam.	Superheated Steam.
Temperature of the Steam	Centigrade . .	164	460	164	500
	Fahrenheit . .	327	860	327	932
Power developed	Metrical B.H.P.	21·4	24·5	44·1	51·9
	English B.H.P.	21·1	24·2	43·5	51·2
Steam Consumption per B.H.P. per hour . .	Kilogrammes per Metrical B.H.P.	21·6	14·1	17·7	11·5
	Lb. per English B.H.P.	48·3	31·5	39·6	25·7
Heat Consumption per Metrical B.H.P. per hour in Metrical Heat Units		14,160	11,270	11,610	9,390
Temperature of Exhaust Steam	Centigrade . .	100	309	100	343
	Fahrenheit . .	212	588	212	649

With the high speeds we have indicated above, a great difficulty arose at first with the excessive vibrations, and care has to be taken to make the wheel of such a section that it shall be strong enough to resist the centrifugal force which is generated. This is so great that on the bucket of a 300 H.P. turbine wheel which bucket weighs 250 grams only, the centrifugal force is 1,680 lb. when the wheel is running at its standard speed. The wheels are made of solid steel of the form shown in Fig. 234, *A* being the half boss shown in section and *B* the bucket.

No matter with what nicety the turbine wheel may be turned and balanced, it is practically impossible to bring the centre of gravity of the wheel exactly into the geometrical centre round which the wheel revolves. This fault causes vibration, which, if a firm shaft were used, would increase with the speed to such an extent that the bearings would quickly be ruined. To obviate this, a flexible shaft is used, shown in Fig. 235. This is shown at *A* and a side view of the bucket at *B*. The stresses on the wheel are so great that in the large sizes the wheels are made without any hole in the centre, and the shaft is made in two pieces bolted on to the boss, as is clearly shown in Fig. 236. Even with the employment of the flexible shaft there are still vibrations increasing with the number of revolutions of the wheel. At a certain speed, however, the vibrations suddenly disappear and the shaft runs smoothly in its bearings.

Flexible Shaft.

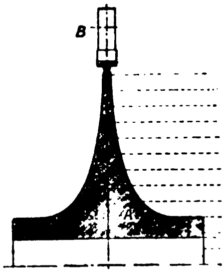


FIG. 234.

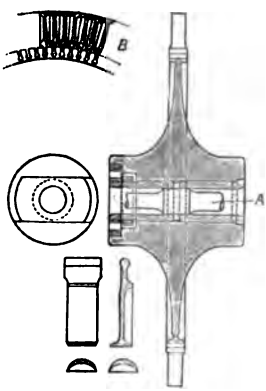


FIG. 235.

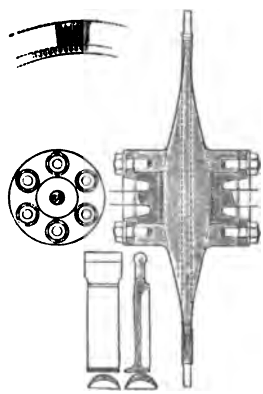


FIG. 236.

Critical Speed.

The speed of the wheel at which this phenomenon arises, is called "the critical speed of the wheel." The explanation is that the wheel at this critical speed takes a new centre of rotation very near to, or at, the centre of gravity, the shaft springing and allowing this to happen.

It is difficult to give a scientific explanation of this pheno-

menon, but assuming, as is very probable, that the settling of the wheel occurs when the number of revolutions is equal to the number of vibrations which the shaft makes with the wheel mounted upon it, the critical speed may be calculated, and it is found to be $n = C\sqrt{\frac{P}{Q}}$

When P=the force required to bend the shaft a certain distance.
Q=the weight of the turbine wheel.
C=constant.

This formula is stated to correspond very closely with the results obtained at actual tests.

The flexible shaft and the turbine wheel are so proportioned that the settling of the wheel takes place very quickly, and the critical speed is $\frac{1}{6}$ to $\frac{1}{8}$ of the standard number of revolutions of the wheel. The turbine wheels are very finely balanced, and the " settling " is scarcely perceptible.

The diameter of the turbine shaft is very small and is, therefore, easily made flexible. The shaft of the 300 H.P. turbine wheel has a diameter of only 34 mm., or $1\frac{5}{16}$ of an inch, and that of the 150 H.P. is only 1 inch diameter, and these are found to be amply large enough owing to the great speed at which they run.

The turbines are generally constructed with more than one steam nozzle; the nozzles are arranged at intervals in a ring in close proximity to the turbine wheel, and receive their steam from a steam chest in the turbine case. The whole construction can best be seen by reference to Fig. 237, which shows a vertical section through the turbine case, and also the gear case, and a plan of turbine case and section of gear case.

The steam is admitted by the stop valve A passing in to the steam chest D and E by the double-beat valve shown in drawing below D, and which is controlled by the governor. The steam chamber E E runs quite round the turbine case, and in this chamber the nozzles A', B, D, and V are fixed; these are shown by inclined lines in the view, but an enlarged detail of the nozzle is shown in Fig. 238.

Reducing Speed Gear. The turbine wheel, a detail of which is given in Fig. 235, bears a small proportion to the whole so far as size is concerned, and is shown at F. The steam passes from one side to the other of the wheel, and then enters the

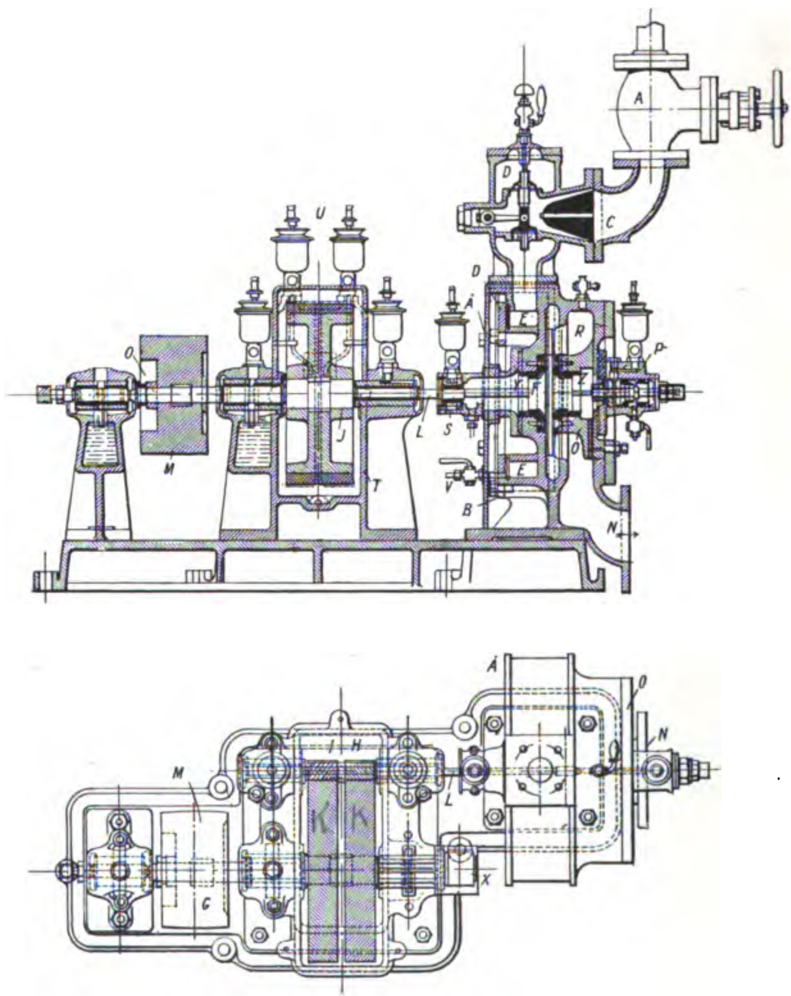


FIG. 237.

exhaust chambers *R O* and passes out into the exhaust pipe by bent opening *N*.

The turbine spindle, with its 20,000 revolutions, passes out at *L* in the plan, and working between two bearings is enlarged at *H*. The enlarged portion after being turned true is cut with fine helical teeth right and left-hand, and gears into the helical wheel *K* at a ratio of 10 to 1, thus reducing its speed to 2,000 revolutions per minute, which also is the speed of the driving pulley *M*.

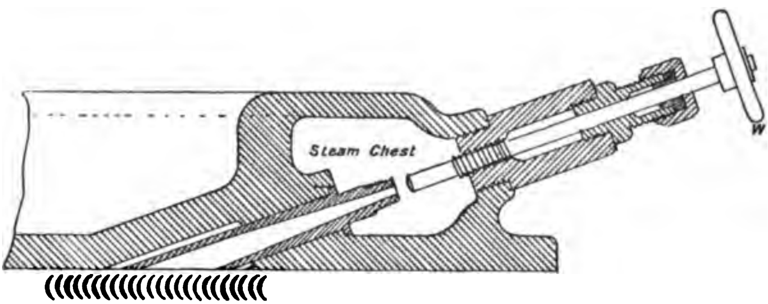


Fig. 238.

Governing. The speed is regulated by a governor controlling a double-beat valve. This of course varies the pressure in the steam chest, and the turbine works most economically with the highest pressure available. In order to maintain this the following method is adopted. Say, for instance, that the turbine runs at full load and that the machine is designed for 180 lb. per square inch, then the admission pressure at the stop valve would be 180 lb. above the governor valve, and should be, and generally is, from 170 lb. to 175 lb. below the governor valve; this depends upon how the nozzles are proportioned. Now suppose that a good deal of the load is taken off suddenly, the throttle valve would be closed by the governor to such an extent that the working pressure below the throttle valve would be reduced to say 120 lb. per square inch. This does not matter if it is only a temporary reduction in power, but if the turbine has to run at this reduced power for any considerable time, it is more economical to close one or two of the steam nozzles then in use. A reference to Fig. 238 will show that this is very simply and easily done by one or two turns of the hand-wheel *W*. As soon as a nozzle

is closed, the pressure in the steam chest would rise because the governor would open its valve a little wider, and still other nozzles should be cut off till the steam chamber pressure rises to at least 170 lb. or within 10 lb. of the boiler pressure, as it is much more economical to work two nozzles at full pressure than four at half pressure. From this it will be seen that this class of turbine as well as others, is most economical and convenient to be used for uniform loads, or at any rate, loads which are not constantly changing.

Invention has been very active during the last few years, and there are now many other forms of steam turbines besides those which have been described. Amongst these are the Riedler-Stumpf, which has its buckets machined out on the periphery of its wheels and is something like a steam Pelton wheel; this has been made up to 2,000 H.P. and gives good results.

The Zolly.—This is a many-stage impulse turbine and partakes something of the character of both Parsons and Laval.

The Curtice.—This turbine is one of the few which works with a vertical shaft, the wheels like a number of Laval turbines being placed one over the other.

The Rateau.—This is a pure impulse turbine and consists of a number of thin steel plates pressed into a slightly conical form to give them stiffness. The buckets are formed by partly cutting and partly pressing a narrow portion of the circumference of the plate into the desired shape to form the buckets or blades. This turbine has had a considerable success.

The Shultz patent, 1900, is of the Parsons type with the high pressure and low pressure sets of blades divided into two conical drums on either side of a central admission. The steam entering the smallest of the high pressure set of blades expands to the largest, and then returns to the centre and again commencing with the smallest expands to the largest and then to the condenser. Both sets of blades, increasing in diameter from the centre, if properly proportioned balance the longitudinal pressure.

The Lindermark is one in which the inventor, formerly chief draughtsman with the Laval Co., Stockholm, transforms the speed energy of the steam into pressure so as to get a lower peripheral velocity.

There is also the Gelpe-Kugel and numbers of others, the practical development of which will be looked for with interest. The long successful use of the Parsons and the Laval turbine has removed them by many years from the experimental stage, and the invention and development of which marks a distinct and important epoch in steam engine construction.

CHAPTER XIV

Design of Details

In designing a steam engine, there are three things which have to be kept in mind at every stage in our progress; they are strength, economy and beauty. The first two are essential, the third advisable, and it will come almost as a matter of course, if the two first are secured. "That which *is* right will always look right," was a shrewd maxim instilled into the author's mind by one of his earliest teachers, the late George Wilkinson, a daring innovater in engine design; and they are true words. In order to secure these things the designer should keep as much as is possible to straight lines and simple curves, especially the former. There is a well-known quatrain which puts the matter in a nut-shell, and though not written in connexion with steam engines, it is equally true whether applied to ethics, politics, or mechanics—

General Principles.

Straight is the line of duty,
Curved is the line of beauty;
Walk by this first, and thou shalt see
The other ever following thee.

Importance of Details.

A steam engine is built up of many parts, and, as a chain is no stronger than its weakest link, so an engine is no better than its worst part.

To any who have a large and general engineering experience the hearing of such expressions as the following is only too common, "Yes, mine is a real good engine, if only it wasn't for that confounded crank pin heating," or, "that governor hunting," "that feed pump failing," or some other detail going wrong. Only a few months ago the writer was called in to advise about a serious breakdown with an engine, and found a broken piston and cracked cylinder, all resulting from

the neglect of the designer to provide suitable relief valves to the cylinder; a small affair, worth only a few shillings to put in when the engine was being made, but the repairs to which cost very many pounds.

Each part of an engine should be designed as if it were the most important part, upon which all depended, as occasionally it may do. The two bolts holding in their place the halves of a marine-ended connecting rod, are a very small part of an engine, and whether made "just strong enough," as they often are, or with a wide margin for contingencies, as the experienced designer does, make practically no difference to the cost of constructing an engine; yet the representative of one of our largest insurance companies informed the author that there were more accidents resulting from the breaking of these two bolts than from any other single cause.

The above is not intended to mean that there should be any waste of material, but always enough and to spare, that every ounce of it should be put in the best place, and that the shape of each part should be such that the necessary work can be done to it with ease and celerity.

The first and greatest thing to be considered, is the interest of the user of the engine; next, that of the manufacturer and the artisans. If the user gets a good engine in all its parts it will be a pleasure and profit to him, for as long as he uses it; but, if it be a bad one, then it will be a source of annoyance and loss until it be replaced by a better one. Again, it is no worse but *better* for the user, that the engine should be easily and economically made, for then the manufacturer can sell it at a reasonable price and still get a fair profit. If the artisan has also been well considered, then he will have ease and joy in his work, and will get much more done, as there will be less strain on his muscles and none on his temper. A good designer can do more to promote morality and prevent profane swearing in a factory, than all the ministers and clergymen in the district.

It would be impossible, as well as futile, to give a formula and definite rule for the strength and proportion of each part of an engine. The broad principle above laid down must be kept in mind, as well as the drawings and descriptions of the various parts of an engine given in the body of this book; but, above all, the designer must make himself, and

keep himself, familiar with the best examples of engineering practice. Just as an artist cannot paint a beautiful face merely by knowing what proportion the length of a nose should bear to the width of a mouth or height of a forehead, so the draughtsman cannot design a first-class connecting rod merely by knowing that it should have a length, if possible, at least three times that of the stroke of piston. The illustrations and drawings in the book are well worthy of consideration, and in the following pages a few typical examples will be given of what to do and how to do it.

Base Plate. Take for instance an engine base plate; this must be strong, true and rigid.

Formerly it was the custom to make base plates quite flat and heavy, and to bolt all the fixed parts of a horizontal engine on to the upper surface, securing them by dowels, steady pins and wedges where required. Amongst these parts would be the heavy guide bars (or slide bars, as they are called) for the piston crosshead.

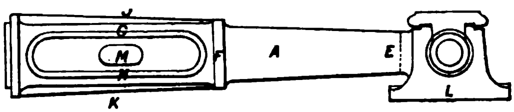


FIG. 239.

Allan Trunk. Some fifty years ago the "Allan" or bayonet trunk bed came to us from America, and was soon after adopted by some of the best makers in England and on the Continent, though a very large number of makers still use the flat bed.

It is not too much to say that for long stroke engines the bayonet bed, as shown in Fig. 239, gives greater strength and rigidity, with only about one-half the weight of metal that is required for the flat bed. In this trunk the metal is disposed as near as can be in the direct line of the strain. The tendency of the strain of the steam and working parts is alternately to draw together and push apart, the cylinder and the main bearing. This strain, so far as the cylinder is concerned, is perfectly resisted, as it is bolted directly on to the end of the bed, and the main bearing forms a part of the trunk itself, all the strain being carried through the straight arm *A*, which is directly in the line of strain in elevation in Fig. 239,

and as near as possible to it in plan (Fig. 240). The arm is further stiffened to prevent bending by the enlargement at *B* (Fig. 240). This arm is made hollow, as shown by the dotted lines, and as cast-iron is much stronger in compression than in

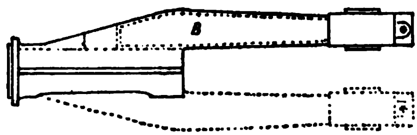


FIG. 240.

tension, the side of the arm nearest centre line should be made about 50 per cent. thicker than the other side. Now this class of engine trunk has to be made both right and left handed; lack of thought and care in the design would compel the making of two patterns and two sets of moulding boxes; but, by making the trunk symmetrical, and dividing the pattern at line *E* (Fig. 259), the bearing end *L* can be turned up or down; and so with a very trifling extra cost we get two patterns out of one and both can be moulded in the same box. The right and left-hand patterns are shown (one in dotted lines) in Fig. 240.

Base Ornament. Many such engine trunks are made with what are intended for ornamental mouldings, as at *F*, *G*, *H* (Fig. 239); these naturally increase the cost of making the pattern, which is only once, and also the cost of making, and more especially finishing a mould, the latter being at least doubled, and this occurs every time a trunk is moulded. And they serve no useful purpose, the severe simplicity of the outline of Fig. 241 being preferable as well as less costly. The mid ribs *J* and *K* are, however, useful, and give extra stiffness to the trunk, which also forms the crosshead guide.

The elongated hole *M* is an important feature, enabling the crosshead-pin to be taken out and put in with the least trouble. The long opening *N* also must be made of such a width that the crosshead can be put in or taken out from the side, as this is often the most convenient way. And here let it be said that the draughtsman in making his design must at the same time carefully consider how each part can best be forged, moulded, tooled, fitted together, and taken apart again, the last not being by any means the least important.

A section through the line *M* is shown (Fig. 242), from which

it is seen that the trunk is bored out to form the crosshead guide. Some makers use flat surfaces for the guides, but this is an expensive method and has no corresponding advantages. The trunk should be bored out in a special tool, which can, at the same time, bore out the seating for the main bearing and

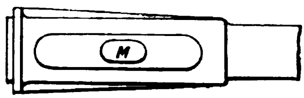


Fig. 241.

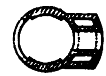


Fig. 242.

turn up the end of trunk for the attachment of the cylinder; thus everything is compulsorily made perfectly in line and square as it should be, without needing any highly skilled labour for adjustment. This form of trunk is used for quite small engines, but its great advantage over the flat bed is most marked with large engines having long strokes.

Mammoth Bed. When in addition to the ordinary fly wheels, extra weights have to be carried on the main bearings, such as the revolving armatures of an electric generator, then a much more substantial form of main bearing and support is advisable, and what is known as the mammoth form of bed is used. This is shown in Fig. 243 in elevation and Fig. 244 in plan. This is suitable for the very largest engines; the bed is in two parts, the main bearing and its foundation from *a* to *a* being in one part, and the trunk guide in another; it is the more convenient for manufacture and transport both by rail and steamship. The question of transport is one which

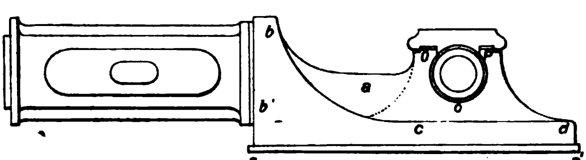


Fig. 243.

must always be considered by the designer, as heavy weights are carried at a much higher rate than light ones, and sometimes cannot be taken at any price to certain parts of the world.

The general design of this form of bed came from America, the special form shown in Figs. 243 and 244 being the author's

modification to make it symmetrical, in the year 1898. This is a very expensive pattern to make, and thus all the greater reason to make one pattern serve for both right and left-hand beds. This is done by making the bed part separate from the bearings, i.e. all the part to the line *b*, *c*, *d* is built symmetrically each side of the centre line ; then the bearing and the bed extension *e*, *f* in Fig. 244 is made, and it is evident it can fit equally well on both sides. Thus the only part which needs duplicating is the small filling piece *a* to the dotted lines, which has to be made right and left-handed. These parts are all dowelled into their places, and being separate from the main part of the pattern, are more convenient for moulding.

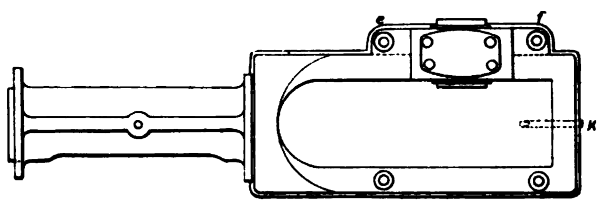


Fig. 244.

A bottom plate is cast in between the inner sides of the bed, and this part forms a very convenient well in which to catch the oil from connecting rod and main bearings, which might otherwise be wasted. At *K* a small pipe is cast in with a screwed end projecting outwards ; a cock is screwed on the pipe, by means of which pipe and cock the oil can be drawn out from the well for filtering and re-use. Another pipe is cast in under the main bearing so as to convey into the bed well the oil from the outer side of main bearings ; this is shown more in detail in Fig. 245. The insertion of these pipes in the mould costs practically nothing, but saves a good deal of trouble and expense afterwards.

These are but small and apparently trivial matters, but the sum of all such like savings in an engine factory makes a very material difference to the profits at the end of the year.

Tooling Mammoth Bed. In a well-equipped factory such a bed would be placed on its side, firmly secured on a true bed, and the bottom part, *a a*, would be tooled by a travelling milling plate with a number of cutting tools which would reduce the casting to a true plane in one operation. At the

same time another milling head would tool the parts *O*, *P* of the bearing ready for the reception of the cap. The bearing cap would be tooled to template in a smaller machine. An air-driven portable drilling machine would drill the four holes required with which to bolt on the cap, and thus by the time that the bed bottom was finished tooling, the cap would be fixed in its place. The boring heads and bars of a suitable machine being at right angles with each other, and adjustable to distance and height, would be set at the same height, and then both the ends of bed *b b* would be bored and faced for the trunk, and the bed and cap for bearing brasses would be done at the same time; and in this manner absolute truth of parts at a minimum cost would best be secured.

Main Bearing. The author's design of main bearing (Fig. 245), has borne the test of some twenty years' use in many hundreds of examples, and has shown itself to be as entirely satisfactory in use as it is economical in manufacture. The main considerations in such a bearing are perfect truth, absolute fit, and easy renewal of the bearing brasses. These are made as shown in Fig. 245 in four parts, *a*, *b*, *c*, *d*, and may be either of gun-metal, or be cast-iron shells lined with white metal, the best mixture for which, in the author's opinion, is tin 85 per cent., copper 10 per cent., and antimony 5 per cent. Nothing can be better than this mixture, though there are many alloys much cheaper, lead largely taking the place of tin, but for an important part of the engine like this, too much care cannot be exercised to have everything of the very best, regardless of the difference in cost; this after all is not much, as the quantity required is small. The method of fixing the alloy in the casing is clearly shown; the casing must be clean and warm, and the small amount of antimony in the mixture, is enough to secure the expansion of the alloy in cooling, thus securing a tight grip of it.

White Metal Alloy.

Four-part Bearings. The shape of the four parts is important; the top and bottom pieces, *b* and *d*, are of the same shape, the two side pieces, *a* and *c*, are shaped differently, but are equal to each other. It will be noticed that the part *c* has wings or projections at *e* and *f* which extend as far as the flanges of the brasses, and prevent the parts *a* and *c* from closing in further than the wings allow. This is done for two reasons, viz. convenience of manufacture and

truth of adjustment. After casting in the alloy, each part is simply planed to the shape shown. The four pieces can then be secured firmly together by broad iron clips surrounding the unturned parts *G* and *H* (Fig. 246), when the four pieces can be tooled, i.e. bored out and turned up with the same facility as if they were all in one piece. The inclined flat surface *K* (Fig. 245) is next planed to a definite angle, and then beyond milling a groove for the oil at *I*, *J*, and drilling and tapping a $\frac{3}{8}$ hole in the centre of the flange of each part, the brasses are completely fitted and finished; thus in two or three days from leaving the foundry a large and important part of the engine can be made ready for the erector.

The two holes drilled in each part of the bearing brasses are in the centre of the thickness of the flanges, as at *L*, and are for the purpose of putting in and taking out the brasses; a long bolt, like the one shown in dotted lines at *L* (Fig. 245), is screwed in the tapped holes by hand, and by this means they are held in position and dropped easily into their places, a rather difficult and dangerous occupation for the fingers without this appliance. Further, by this means, if by any chance it is found necessary to remove the brasses while the shaft is in position, it can be easily done. The wedges *K* and its companion are dropped down into the cavity *M*, then when the cap is removed, the top part *b* is easily lifted out by the lifting screws; these latter are then screwed on to the part *C* and its opposite, and the wedges being dropped out of the way, the parts *a* and *c* can be turned over till the screws come to the top, when they are lifted out; then if the shaft be lifted ever so little by a jack or wedges, the bottom piece *d* can be removed in the same easy manner.

Bearing Adjustment. We next come to the adjustment; this is by the four wedges, one of which, *K*, is shown. Some makers who have adopted this form of bearing, have gone to more trouble and expense in tooling the inside of the casting behind the wedges, than the whole cost of tooling the bearing as described. The method which careful thought suggests is shown in the section; where at *N* is a rounded projection which may run across from side to side, and the height of which is on the centre line. The wedges are shown rounded in a direction at right angles to the swell on the casting; thus at *N* they touch at the intersection of the two curves.

allowing for a movement in any direction. The planed surface *K* therefore adjusts itself perfectly to the planed part of the bearing. Thus with practically no expense a perfect fit is secured. When first put into place the brasses fit their seating and the shaft, and no adjustment is needed; the wedges are drawn up tight, this drawing all parts together quite tightly, but owing to the projection at the wings, it is impossible to make the bearing grip on the shaft and become over-heated. If properly made and of right size, such bearings will last for many years without any further adjustment; and, when any is at last required, all that is necessary is to take out the side pieces *a* and *c* and remove by a fine file a very little off the four wings of each part, and then replace and screw up the wedges.

The two lugs, *T U*, at the end of the bearing cap which grip the main casting, are not for the purpose of giving any strength to the bearing, which is amply strong without them, but they serve to fix and retain the cap in its place, thus securing a perfect fit for the brasses. While the bearing is being bored out, two strips of iron about ¼ inch thick are placed between the cap and bearing, thus leaving the gap shown for screwing down after all is put together. The facing, *R S*, and the surface of the cap are the only visible parts which are tooled or need tooling. The surface of the cap being the only part visible when all is together, is alone polished. It must be remembered that the best engineering does not consist in the number and extent of the polished parts, but in the ample strength and perfect truth of the parts which fit and do the work.

If all straight lines are really straight and all curves true, but little polish is required to produce beauty; but what polish there is must be of the best.

The tube for the oil should enter the top brass as shown and terminate in the grooved channel by which the oil is distributed over the whole length of the bearing. The channel is not deep but wide, as seen at *b*, and the sides, therefore, form a small angle with the shaft and this spreads the oil over it instead of scraping it off, as they would do were the side approaching a right angle. All the parts of the bearing fitting closely together, there is no outlet for the oil excepting at the ends of the bearings, and it slowly exudes between these and the collars of the shaft; that coming out by the end *V* falls directly

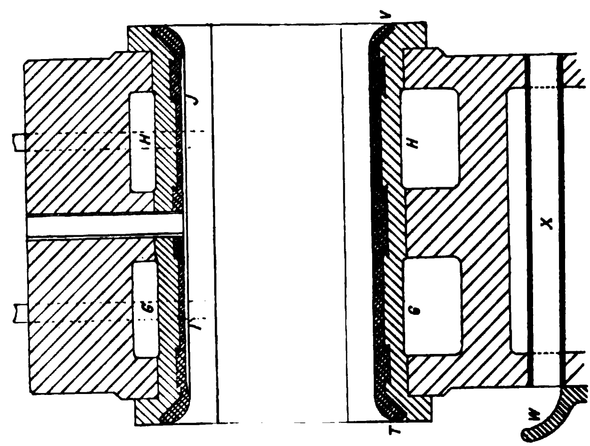


FIG. 246.

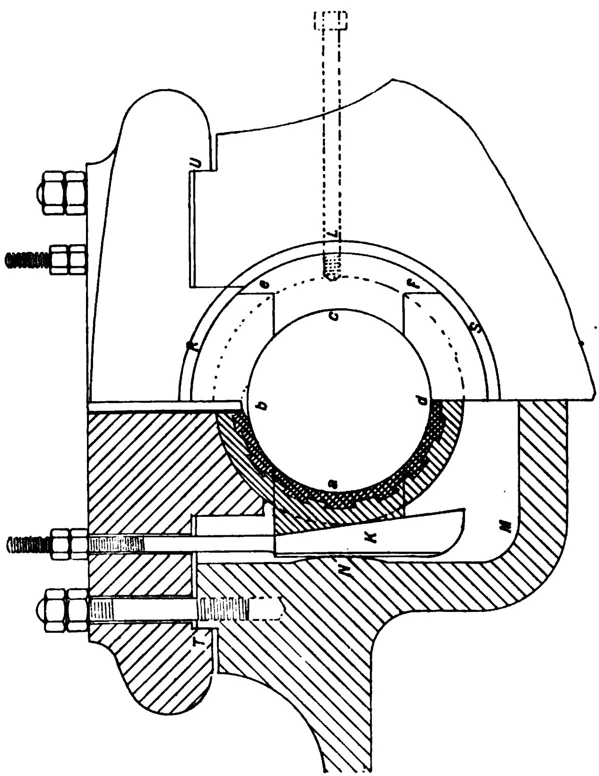


FIG. 245.

into the oil well, and that coming by the end T falls into the channel W prepared for it, from whence it is taken by the pipe X (previously mentioned as being cast in) to the oil well.

The main bearing of many engines is the part which gives most trouble. One constructed as Figs. 245 and 246, and properly oiled, gives absolutely none.

Main Bearing Area. A good rough rule, and one that is easily remembered, is for long stroke horizontal engines, to make the diameter of the main bearing half that of the cylinder, and its length double its diameter.

Of course, this varies with the pressure, and the best way is to make it of such size that the pressure on it shall be not much more than 150 lb. per square inch.

Crosshead. Another part which often gives trouble and which demands much care in its design, is the piston-rod crosshead and pin. There are a great many ways of making this connexion between the piston-rod and crosshead. Some makers forge a block on the end of piston-rod, make that into the crosshead, and fit it with bearing brasses and a cap; this necessitates a forked end to the connecting-rod, and is far from a satisfactory method, the piston cannot be so easily fixed or taken out, and the forked end required for the connecting-rod, making it much more expensive, both in forging and tooling.

The way preferred by the author is to fix the piston on its rod when it is outside the cylinder; it is then easy to test the freedom, fit and elasticity of the piston rings; then altogether it can be placed in the cylinder, and the piston-rod secured to the crosshead by a cotter.

The end of the piston rod and the base of the crosshead are made taper about 1 in 16, and, when cottered up in this way, is solid and secure, as well as economical in cost. The lock-nut *a* behind the crosshead boss, in Fig. 247, is not for making the attachment any firmer, but is merely for convenience and ease in removing the piston-rod when required; without this nut it is a difficult and tedious process, as it would have to be driven out from the crank end, and the connecting-rod would have to be taken off for the purpose; there is, further, the risk of bruising or injuring the end of the piston-rod, as this gets very tightly fixed. With the lock nut, however, the removal is quite easy; a quarter turn of the nut effects the

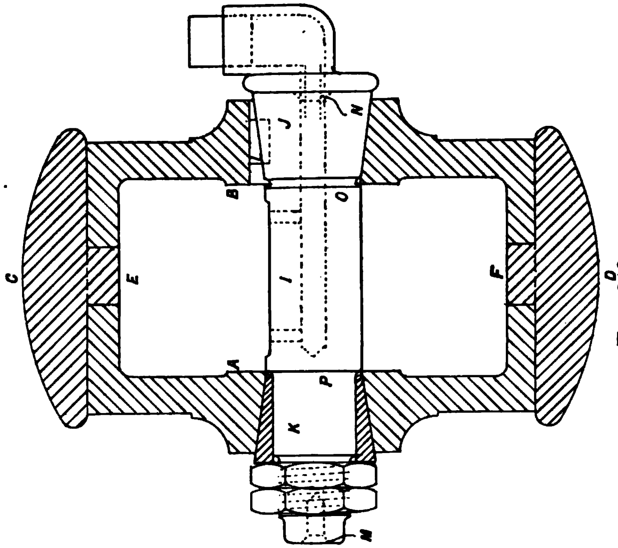


Fig. 248.

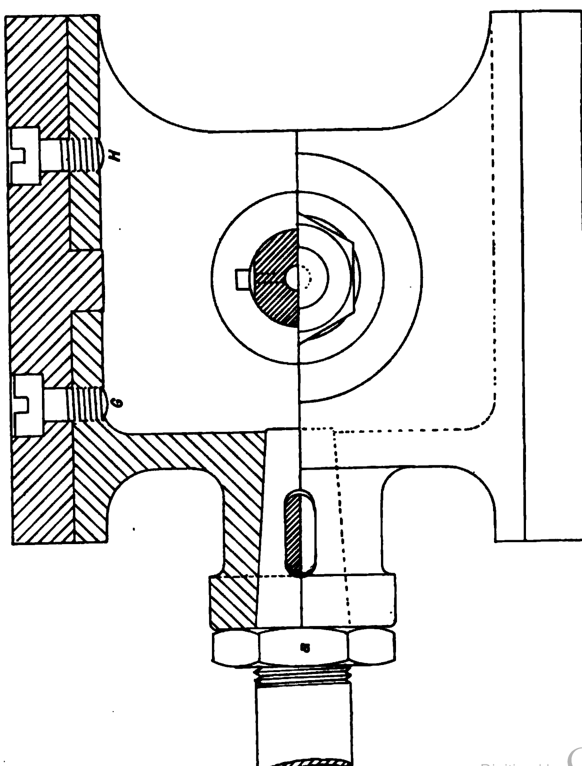


Fig. 247.

removal, and secures blessings instead of curses on the head of the designer. The crosshead itself is made of homogeneous cast steel and is worthy of the student's careful inspection. To begin, the exact shape of the inner part or chamber, is made by a core which leaves all the inner part the exact size required with the exception of the two facings *A* and *B* (Fig. 248), which project from the inner surface sufficiently to allow of their being truly tooled and being still $\frac{1}{16}$ to $\frac{1}{8}$ (according to size) clear of the untooled surface. The boss for the piston-rod is first bored out, all parts tooled truly from this as a centre.

Cast steel, though best for strength, is not nearly so good as cast-iron for a rubbing or wearing surface; in fact, two cast-iron surfaces, if truly made, well oiled, and of sufficient area to prevent cutting, will wear practically for ever; they soon take a bright hard surface, and every year's use seems to make the surface brighter and harder.

It is for this reason that the two plates *C* and *D* are made of hard cast-iron and fixed as rubbing surfaces on the crosshead. If connecting-rods could always be made from three to four times the length of the stroke, there would be no excessive pressure on the guides, owing to the small angle they would then make, but from lack of space or other reasons they have sometimes to be made from two to two and a half the stroke, and then there is great pressure on the guide surfaces, and it is good practice to line the slide plates, *C* and *D*, with a surface of white metal like that used for the bearings. From 50 lb. to 100 lb. per square inch is the limit of pressure on the surfaces.

It is the practice of some makers to insert adjustable wedges or screws between the body of the crosshead and the slide plates; this is a practice which cannot be approved, and is extremely dangerous. The author has known not a few cases where the injudicious adjustment of the wedges or screws has made the crosshead too tight and resulted in most serious damage to the guiding surface. A solid good fit is much to be preferred to any adjusting apparatus; once made (and it is very easily made with such a crosshead as shown) it will last for many years; after which, if any adjustment be required, it is best done by inserting the very thinnest plate of sheet metal or even of thin tracing paper between the crosshead and the plate, and screwing up again and fitting in.

The slide plates are fixed by means of the projecting bosses

or pins *E* and *F* which fit into holes of the same size in the crosshead, and the plates are further held in their places by the screws *C* and *H* (Fig. 247). The screws serve to hold the plates in place while being turned up ; when turned on a true mandril, the crosshead is absolutely true and central, and needs no fitting or adjustment by hand ; the side edges of the plates are rounded off as shown, which makes the easiest and best finish.

Crosshead Pin. The important matter about the crosshead guide is the pin *I* (Fig. 248), on which the connecting-rod works. This pin takes the whole strain of the engine and is more likely than any other part to get loose and knock.

If it be merely a parallel pin, driven in, as is the case with very cheap engines, however well it may fit at first, it is sure to be loose and knock after a few months' hard work. In order to prevent this they are frequently made with two tapers of the same degree (Fig. 249), but naturally of different sizes. This

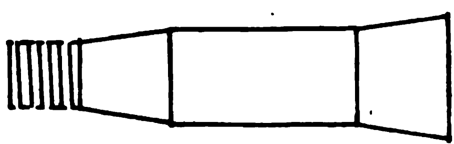


FIG. 249.

is better, but not perfect ; it is a difficult and delicate operation to get both tapers exactly right, so as to be sure that it is an equally tight fit on both sides of crosshead, as if one taper be in the least degree too big, the other must be loose and will knock. The pin designed by the author about thirty years ago, is that shown in Fig. 248. The crosshead is bored out equally on each side, i.e. to the same taper and the same size ; thus it is either right or left-handed. The taper advisable is 1 in 8, as shown. The part *J* of the pin fits the tapered bore ; the part *K* is quite parallel and is surrounded by a tapered ring. The ring, after being turned, has a slot or nick cut in it, allowing it to spring in or out very slightly. Then when put into its place, as shown, and the nuts tightened up, the ring fits tightly like a conical wedge between the crosshead and the pin, and holds it as firmly in position as if it were all in one solid piece. A small key *L* is let into the head of pin and fits into a slotted key-way, merely to hold the pin in the correct position ; there

is no strain whatever on the key. The upper part of pin near *I* is flattened to form a receptacle for the distribution of the oil. The pin is oiled from the brass oil cup (Fig. 249A), which is fixed by two screws on to the end of the pin and conveys the oil through the centre hole and through the two vertical holes to the flattened part of pin, where it is distributed over its whole surface. At one end of the guide trunk an oil cup is fixed, which is adjusted so that it drops one drop each time it is touched by the licker *a* on the pin oil cup. By taking this oil at the end instead of in the centre of the stroke, it is never splashed away, but always dropped into the cup and used. A pin and set of nuts and cam cones if made in the usual way on an ordinary lathe, would take the greater part of a day to turn, and would not then be a dear job. If made in quantities out of a long bar of steel in a hollow spindled lathe, with multiple tools in a revolving capstan, it can be accurately turned in less than an hour. The hollow cones, if less than 1¾ inch diameter, are made out of thick iron tubes, bored, turned and cut off at one operation. The larger ones are quite good, made out of cast-iron tubing. There are two or three other points about this pin which should be noted by the student. Such a pin should be hardened, and then ground again to an absolutely true surface in a suitable grinding lathe between dead centres, the proper centres for which are shown at *m* and *n*.

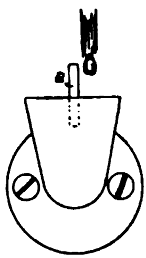


FIG. 249A.

An emery grinding wheel cannot grind perfectly up to a line or shoulder, it must go a little, if ever so little, over ; hence the groove turned in the pin at *O*. There is no need for a similar groove at *P*, as the curve is rounded and the cone does not go quite up to the shoulder. Again, with a fine threaded nut, it is rather difficult to start it truly on the thread, there is always a risk of getting it across-thread and thus making a mess of it ; this is quite avoided by leaving a plain part exactly the size of the smaller diameter of the threads in the nut, and as long as the thickness of the nut, this holds the nut perfectly true, and it runs on to the threads with ease and certainty. These are apparently small and trifling matters, but it must never be forgotten that "trifles make perfection, and per-

fection is no trifle." It is the designer's duty to think out everything, the making, the use, and the adjustment and repairs, and to design every part so that it can not only be easily made right, but that it cannot be made wrong. As there is but little motion on the crosshead pin, it may work with a load of 1,000 lb. per square inch, and if of hardened steel up to 1,500 lb.

Pistons. The piston of an engine is an important detail, so much so that some engineers devote themselves to the manufacture of pistons only; and many of these are of high excellence. There are three principal types of pistons. The simplest form is the solid block with grooves three or more in number, in which grooves, steel rings of small section are fitted; these rings are formed by bending drawn steel bars into a circular shape. The rings spring into the grooves, and for quick revolution engines, these pistons are probably the best that can be used; these are known as "Ramsbottom" pistons.

Another form is made of cast-iron rings turned inside and outside to fit the cylinder, and are pressed outwards against the walls of the cylinder by steel springs. These springs should be very elastic and sufficiently strong to force the piston rings outwards with a pressure equal to that of the steam forcing them inwards, and no more. These pistons, of which there is a great variety, are good and reliable; their principal defect is owing to the fact that they have to be tight enough for the initial pressure of steam, which is high, and are, therefore, unnecessarily tight for the constantly diminishing pressure as the steam expands.

Another type of piston is that which is steam packed and has no springs, a half-section of such a piston is shown in Fig. 250. *A* is the end of the piston-rod, which is enlarged to take the piston, the taper of this point should not be less than 1 in 8 for pistons over 12 inches in diameter, otherwise they have a tendency to get loose and cause a knock at each stroke of the engine. *C* is the piston body or block, and it has a few strengthening ribs between the boss and circumference, one of which is shown at *C'*. *D* and *E* are the piston rings; two pairs are shown, though sometimes one broad one is substituted for the pair. With these and the ring *F* and the cover *G*, we have practically a double piston.

The rings are turned to the exact size of the cylinder after having been first made somewhat larger; then they are cut at one point, the ends closed tightly together and finished to size; this gives a little spring to the rings, but it needs to be very little. They are made of the very best cast-iron, as tough as possible, and as hard as can be, consistently with turning. In putting the piston together, the first pair of rings *D* is put on, then the ring *F*, afterwards the second pair of rings *E* and the cover *G*, the whole being tightly and firmly held on by the nut *H*, a recess for which is made in the cylinder back cover. The front and back surfaces of the piston should be turned; doing this serves two purposes, it removes the outer skin and discovers any flaws which might lead to the piston's rejection, and it makes a perfectly true surface, thus permitting a smaller clearance at each end of cylinder.

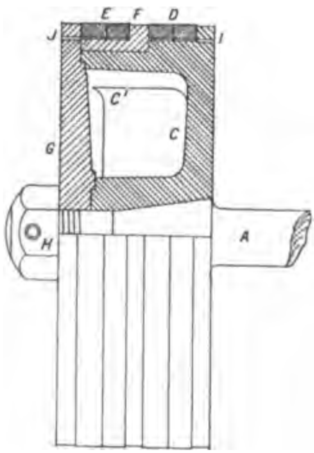


Fig. 250.

The piston body is of cast-iron and must be strong enough for all the strain to which it is normally subjected; but, far from being excessively strong, it should be the weakest part of the engine.

What should be Weakest Part.

If the cylinder be fitted, as it should be, with automatic relief valves, the steam pipe be properly drained, and the water in the boiler kept at a reasonable level, the piston will come to no grief. But in spite of all these precautions, a time may come when a foolish or careless driver may screw down the relief valves till they become useless, or he may neglect his water feed, and fill up the boiler till water comes out with the steam, and once in the lifetime of the engine, the cylinder may be flooded when the engine is going at full speed. Then something must happen. The arms of the fly-wheel may give way; the engine bed may break in two; the connecting-rod may buckle or break; the crank-pin be sheared off; the cylinder covers go, and, perhaps, a part of the cylinder with them; or, the piston

may give way and break up, and this is what should happen rather than any of the other alternatives. When a piston breaks, the engine soon comes to a standstill, meantime no great damage has been done; the piston rod plays about among the débris, but, as there is plenty of room for it, generally no further harm is done to the cylinder bore. Such an accident is easily repaired, and the cause of it investigated and removed.

Every part of an engine should be so proportioned that it will stop rather than break, but this is only when the cause of stoppage comes on gradually; any *sudden* stoppage from whatever cause, must break something.

The method of action of this piston is very simple; in the piston body at *I*, and in the plate at *J*, there is shown a small hole, one of 3 to 6, according to size of piston, arranged at equal intervals around the circumference. A very small space is left between the piston rings and the body, and the small hole communicates with this space. When steam of high pressure is admitted into the cylinder, it immediately enters the holes and fills up the space, and this presses the rings against the walls of the cylinder, making a perfect steam-tight fit. Then, when the cut-off takes place, the pressure falls as the steam expands, and thus, while the piston is always steam-tight; there is no unnecessary friction, but only exactly as much pressure as is needed. Owing to the midfeather *F*, the admission steam cannot pass to the exhaust side of the piston. This piston is always just tight enough, and never too tight.

The Excentric and Strap.

There are a few things about excentric straps which are worth attention, though the construction seems so simple. The ordinary method is to make the excentric rings in one piece, turn and finish them, and then cut them in two pieces by slotting along the line *A* to *B* (Fig. 251), packing pieces being then inserted between the two lugs, and all screwed tightly together again; this seems simple and right enough, but a strap so made is never quite true after having been cut in two pieces, owing to the springing of the metal, and it may need a good deal of work to bed it properly on its sheaf. The method of manufacture designed by the author is as follows:—the pattern of the strap is turned up true, and then cut along the line *A* to *B*, and ¼ of an inch inserted between the lugs, thus making the dimen-

sion *a* to *a'* $\frac{1}{4}$ of an inch greater than the dimension *b* to *b'*, and, of course, the castings will have the same difference.

The casting is placed on the table of a vertical lathe, top side of the mould uppermost, and the ring *C* is turned. If there be any blow holes or such-like imperfections in the mould they will always be on the top side, and then be discovered before any great cost of tooling has taken place; when, of course, the casting will be sent back to scrap. If sound, the surface *C* will be finished, as all other parts are sure to be sound. A true surface having now been obtained, the holes for the coupling bolts *f* and *f'* are drilled to template; they are drilled

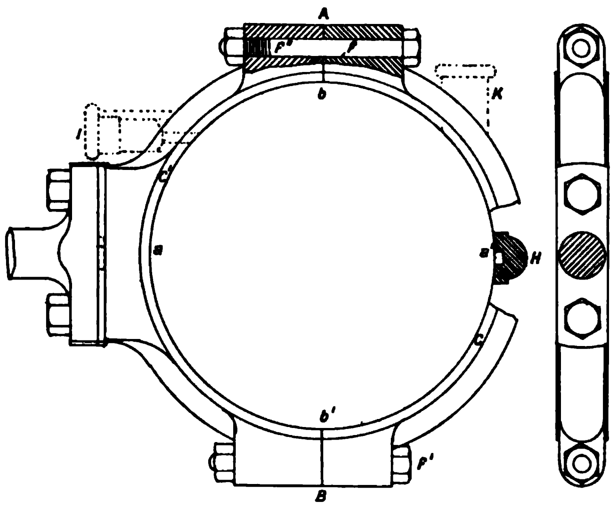


FIG. 251.

as far as *f''*, an easy fit for the bolts, i.e. about $\frac{1}{100}$ of an inch larger; and from *f''* to the end, a tapping size for the bolts, they are at the same time faced truly on the ends of the lugs.

The strap is next cut into two parts along the line *A* to *B* by means of a milling disc $\frac{1}{4}$ inch wide, which not only makes a true cut, but leaves two perfectly true surfaces. The casting now takes its restful position; the bolts are inserted, all is screwed tightly together. It may here be noticed that, by using a portion of half of the strap as a nut *f''*, a lock nut is saved, and one spanner only is needed to put the strap together instead of two, a great saving of time and trouble.

The turning of this strap is now finished from the true surface primarily obtained ; the shape is that shown in section at *H* in Fig. 251, the groove formed serving to hold the strap truly sideways, and also to form an oil well, which is in communication with one or other of the oil cups, shown in dotted lines at *I* and *K*, in accordance with a vertical or horizontal position of the strap. A projection on the otherwise flat sheaf, fits into the groove sideways, but does not go quite to the bottom, leaving a space for oil for lubrication. A strap so made is not only exceedingly economical in manufacture, but is one which works well from the beginning without any hand adjustment; and, if properly oiled, will last the life of the engine.

Slot Link. We have space for only one more illustration of designs, and we will take a slot link, such as is used as part of an expansion gear.

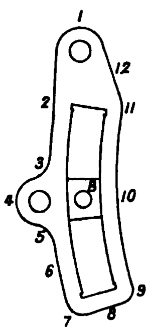


FIG. 252.

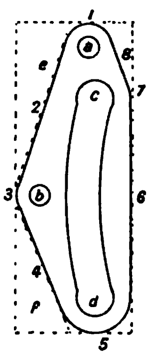


FIG. 253.

A very ordinary form of this link is shown in Fig. 252. This is first forged roughly to size, then planed on each side, the holes drilled, and the slot cut out, after drilling one large hole for the beginning of the slot; this is cut out in a slotting machine, sometimes with an apparatus for making a true curve ; but, more frequently, the slotting tool is adjusted by hand to the curve. The four small spaces at the corner of each end are for clearance for the end fit of the block. The outside is slotted or milled to the form shown, there being 12 different surfaces, numbered consecutively in the drawing. The slide block is done in a similar manner, the two curved surfaces of the slot in the link, and the inner and outer curves

U

of the slide block being difficult and tedious operations, and needing a good deal of time to finish by a highly skilled artisan. The link is then hardened and, generally, slightly distorted in the process. Again it is a difficult and tedious operation to reduce the link to truth and make the block work truly and easily without any shake.

A slot link for the same purpose as designed and constructed by the author is shown in Fig. 253. A piece of steel of the width of one link and the length of two, is sawn off from a bar of the right section; so there is no forging. It is then planed truly on both sides and reduced to the exact thickness required $+\frac{1}{100}$ of an inch. The bar is then sawn in two, thus making two links; these are cramped together and the four holes *a*, *b*, *c*, *d* drilled through both to template. The pieces *e* and *f* are sawn off, through both at once, by the band saw to nearly the exact size; this sawing is much less costly than forging, and the pieces sawn off are worth something. The two links are now slotted or milled to the exact shape shown; the shape is better and stronger than the one shown in Fig. 252 and costs much less in tooling round the edges, there being a smaller number of surfaces to tool, i.e. 8 as against 12, and they are of a simple shape, there being no under-cut.

The two links are now bolted by the four holes, one on each side of a radial arm, which moves vertically between two end-cutting milling tools. The arm being set to the required radius, the curved slot is cut out perfectly true and well finished, and $\frac{1}{100}$ of an inch below size; the clearance at each end is got by drilling the holes *c*, *d* $\frac{1}{10}$ of an inch larger than the width of slot.

The link is hardened and possibly distorted as in the other case, but an emery wheel grinding machine reduces the link in a few minutes to a flat surface on each side and to the exact width, the $\frac{1}{100}$ extra thickness having been left for that purpose, and an emery cylinder rapidly revolving in the slot, while the link travels on a radial arm, speedily produces a perfectly true surface on both sides of the slot; by this means the link has been truly tooled and finished without any hard work by the skilled artisan, and is delivered to him as a finished article to be used in building up his expansion gear.

The slide block is made in a still more simple manner. A long bar of steel is taken six and a quarter times as long as the

radius of the link ; this is bent into a ring which will be double the radius in diameter ; 4 holes are drilled at equal distances and tapped to take a screw, and these are used to fasten the ring on the surface plate of a lathe, when one side of the ring is turned up. The ring is then turned round, being packed off the plate by 4 washers of equal thickness, and then turned on the other 3 surfaces ; it is then set out for the holes, as shown in Fig. 254, which shows a portion of such a ring after the holes

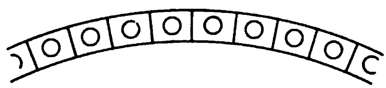


Fig. 254.

are drilled ; the ring is sawn along the lines shown, each piece making a perfect block of exactly the right radius within and without, and without the skilled fitter, to whom it is delivered like the link, finished ready for use. Each ring makes many more than are required, but these are put into store like other small parts which are more cheaply made in quantities, to be used as required.

The perfection of fit obtained by this method is much greater than can be got by hand work ; thus the user of the engine is benefited, the cost also comes out much less ; so the manufacturer also adds to his profit by thoughtful designs.

In these days of keen competition the highest intelligence is needed, and in some manufactures the neglect, or adoption throughout of such methods as we have described, makes all the difference between success and failure.

CHAPTER XV

Examples of Various Types

In the preceding chapters we have dealt with general principles and have referred to only a few types of engine. In the present chapter we shall place before the student illustrations of many different types, leaving out, however, mining and hauling engines, also reciprocating marine engines and locomotives. All these classes are of so great variety within themselves as to require a separate treatise and the last-named is also familiar to all. Of these we mention we have space for only a very brief description, and we begin with the smallest,

Vertical Engine. viz. the vertical single cylinder steam engine; a good example of this is shown in Fig. 255. This type of engine is made from 1 H.P. upwards, but seldom above about 20 H.P., and it is an indication of the great advance in engine design and manufacture that even small sizes down to a 6-inch diameter cylinder are fitted often with automatic expansion gear as shown in the illustration. The engine is frequently combined with a vertical boiler on the same base-plate.

Portable Engine. The class of engine most widely used in all parts of the world by agriculturalists and contractors, when requiring often large power for temporary use, is the portable engine erected over its boiler, and the whole carried on wheels such as is shown in Fig. 256. This is a good example of one of the most modern types. The engine is carried on wrought steel supports, which are riveted to the boiler, and the cylinders are connected to the main bearings by strong steel bars, which not only take all the strain but carry all the motion. There is no type of engine which has had more constant and careful thought bestowed upon it than the portable. Its birthplace was England and the principal seat of its manufacture is in Lincolnshire, but the type has been

copied and is now manufactured in nearly all civilized countries in the world. Such engines are made both high pressure and compound, and in sizes which vary from 4 H.P. to 200 H.P.

FIG. 255.

The one shown in the illustration is a double cylinder high pressure engine.

Robey or Undertype. Another type of engine almost as well known and widely used is that known as a semi-fixed or undertype engine, and is shown in Fig. 257. This engine was designed

and patented by the author in 1870 in the joint names of the manufacturer and himself, and for many years was known as the "Robey engine," Robey & Co. being the sole makers. A diligent search, however, discovered that previous to the date of the patent, one or two locomotives divested of their

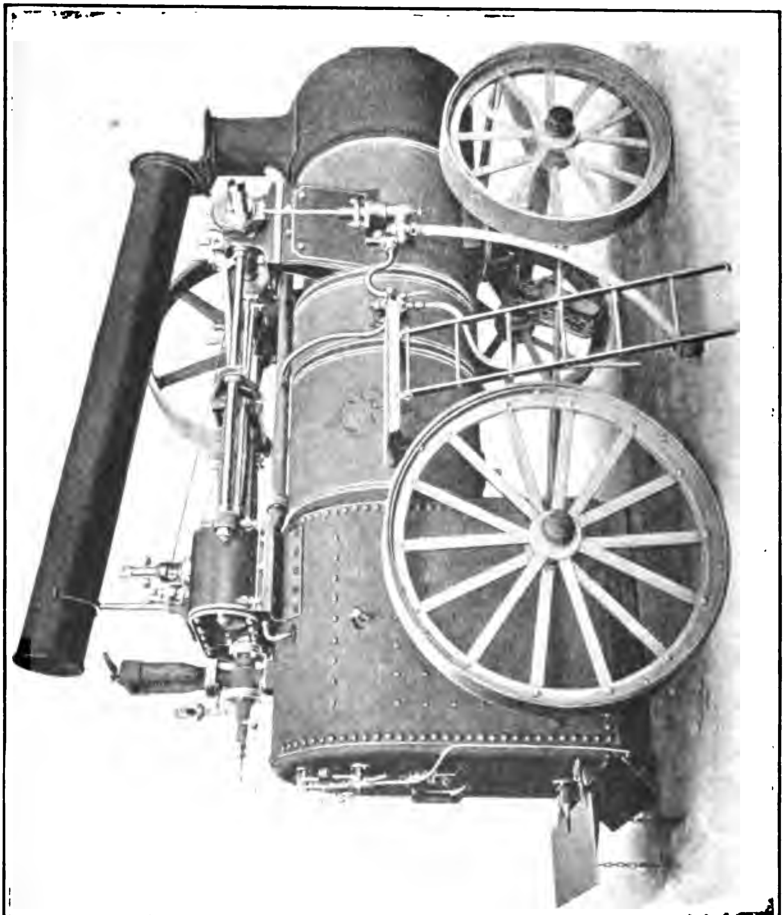


Fig. 256.

travelling gear had been used for driving machinery and hence the patent became invalid, and within a very short time the design was adopted by nearly all the principal manufacturers in England and by many on the Continent. There are few types of engine more widely made and used than this. It

can be delivered, installed and set to work within a few hours, and is made in sizes up to 400 H.P.

The horizontal type of engine has been already illustrated

Fig. 257.

when we dealt with valve gear, in chapter VIII, Figs. 125 and 142 showing very good examples of single-cylinder long-stroke engines. We now show in Fig. 258 a compound Corliss engine by one of the best makers, Messrs. Hick, Hargreaves & Co.

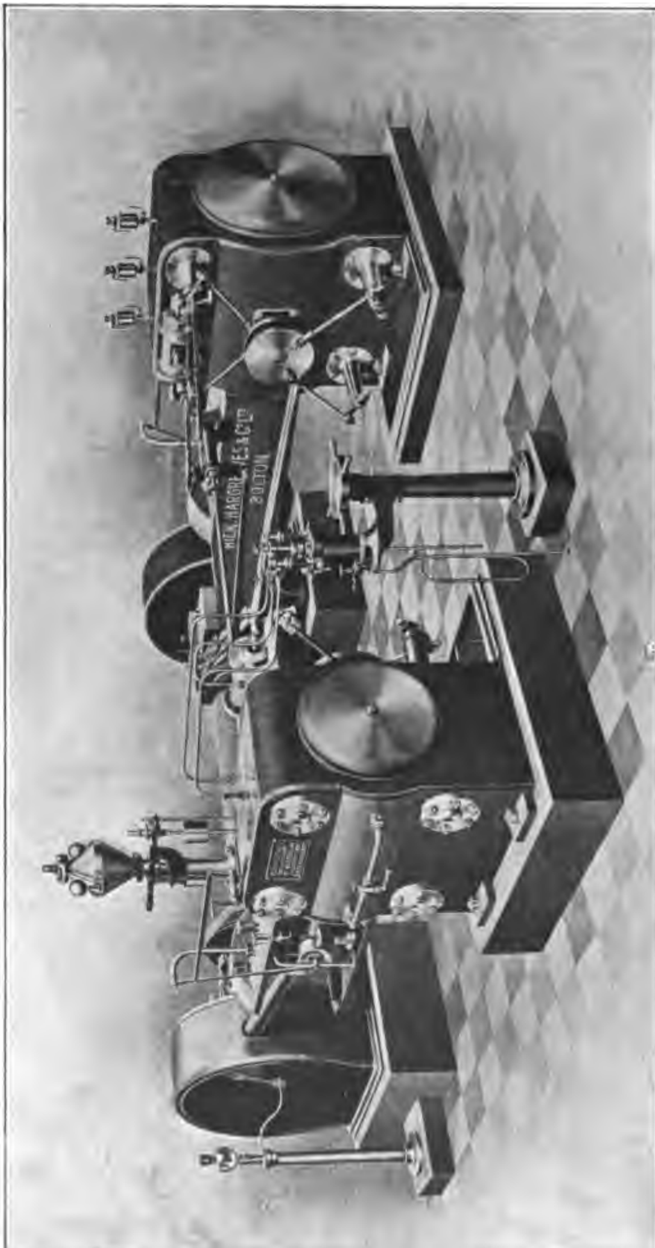


Fig. 258.

The fly wheel, which is grooved for transmitting the power by cotton ropes, is not shown, but its omission enables the details of the engine to be better seen.

Engines of this class frequently carry electric generators on the same shaft as the fly wheel, and a good example of this combination is shown in Fig. 259.

Triple-Expansion Horizontal. This illustration shows a triple-expansion engine by Robey & Co. It has mammoth beds and is fitted with the "Richardson" governor and trip-expansion gear described in chapter VII. It will be noticed that the high pressure and intermediate cylinders are placed tandem with each other on one bed, and on the other is the low-pressure cylinder and the condenser, the latter being hidden by the intermediate cylinder.

This engine is of a type largely used for generating electricity for lighting and power purposes, and the generator for that purpose is seen mounted beside the fly wheel on the main-shaft. An extension to the mammoth bed is made which carries the fixed part of the generator. The engine illustrated works with a steam pressure of 200 lb. and is of 2,000 H.P., but engines of the type of Figs. 258 and 259 are built up to very much higher power than this. Engines of this type working with steam of 200 lb. pressure and condensing use only 10·5 lb. of steam per 1 H.P.

While, taking all things into consideration, horizontal engines of the type shown are probably the best that can be used, being exceedingly economical in fuel, easy to manage and very durable, yet there are circumstances when owing to the space they occupy such prime movers are inadmissible.

Vertical High Speed Engines. Of late years vertical direct-acting engines have largely come into favour; these have generally much shorter stroke and revolve at much higher speeds. There are many varieties of the vertical engine, one of the simplest of which is shown in Fig. 260. These engines are very stoutly built and every working part has very large wearing surfaces; thus they can work continuously at a high number of revolutions and under great pressures. A small engine of this type with a steam cylinder 6½ inches diameter and 6-inch stroke can run up to 500 revolutions per minute, and with a boiler pressure of 100 lb. can develop 20 H.P.

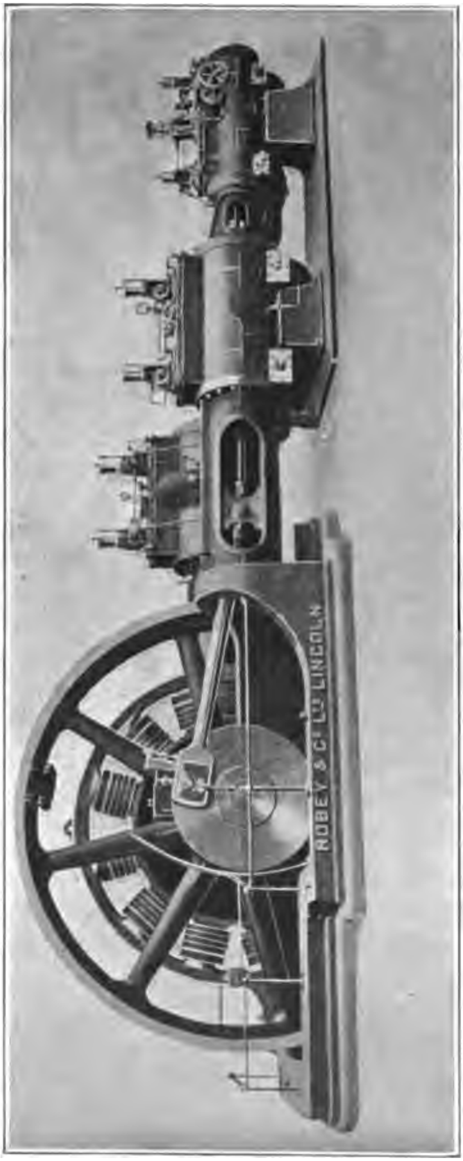


Fig. 259.

Lubrication Drop Feed. Special care has to be taken with the lubrication of this type of engine, and all the parts are oiled from one box fixed just below the cylinder, the various pipes being so arranged that they discharge a single

drop of oil at frequent intervals, the whole of which is collected in the base plate and filtered and used over again. When at work an oxidized sheet steel apron or casing is placed in front

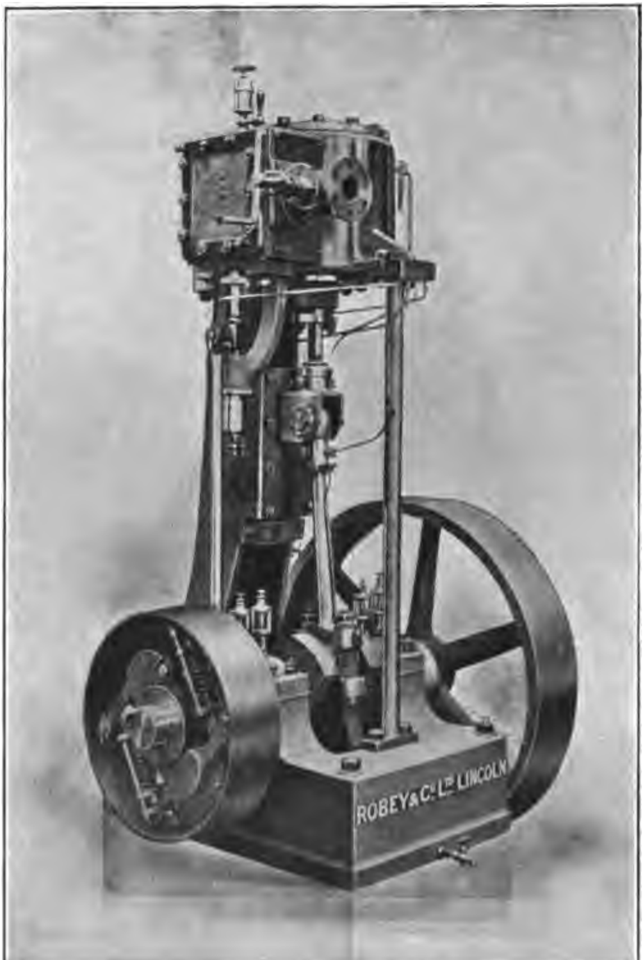


Fig. 260.

of the balanced crank and also around the disc governors which controls the cut-off of a single slide or piston valve. The screen is removed in the illustration so as to show the working parts better. Such engines are made both simple,

compound and triple expansion, each cylinder, as a rule, having its own crank.

Splash Lubrication. Another form of lubrication for high speed engines is that known as "splash" lubrication. With this form the working parts of the engine are all enclosed

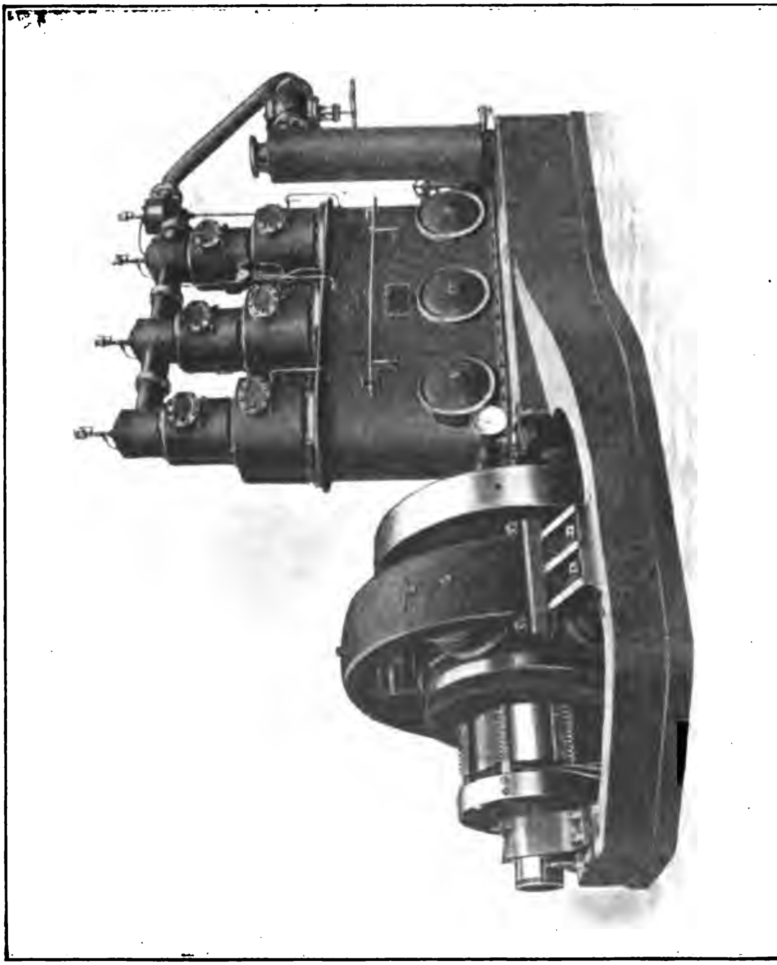


FIG. 261.

within an oil-tight casing which is partly flooded with oil; the crank and connecting-rod just dipping into the oil at each revolution, it follows that the oil is splashed over all the working parts and falls back again into the well. Some slight trickling

of water generally takes place through the glands, some of this falls into the oil well and, mixing with the oil, forms an emulsion which is even better than oil as a lubricant.

The Willans' Engine. Amongst the engines which has this form of lubrication is the well-known Willans' engine. These engines have certain specialities of construction and design which are of great value, amongst the most important of which is the central valve. The Willans' engine has a very short stroke in proportion to its power, and hence it can make a high number of revolutions without an excessive piston speed. All the working parts are enclosed in a cast-iron chamber having doors with oil-tight joints in front. A modern form of this type of engine is shown in Fig. 261 ; the engines have three sets of two cylinders or six in all. The low-pressure cylinders are fixed on casing, on a prepared facing which centres them truly, and the high-pressure ones are fixed above, the steam entering in each case from above and being lubricated as it enters. For very high pressure of steam the engines are made triple expansion, the three cylinders being placed over each other. All the pistons in each set are connected to one hollow trunk. This trunk increases in diameter in proportion to its cylinder and serves as a steam chamber and distributing face for the steam, having ports in it which are alternately open and closed by a service of circular valves working within it. This central valve is worked from the same crank-pin as the connecting-rod, but of course the connecting-rod pin and the valve shown are excentric with each other.

All the cylinders are single acting, the steam pressing only on the upper surface of the piston. There is thus a constant downward thrust of the connecting-rod on to the crank, and only the top half of the connecting-rod brasses are used. To insure this constant pressure during the exhaust stroke an extra cylinder and piston is used ; this takes in air during the steam stroke and compresses it during the exhaust stroke. There is no loss of power resulting from this air compressing, as all the power given to it, is returned to the crank in the downward stroke. There is, however, a slight loss in efficiency due to the friction of the extra parts. It results from this arrangement that such engines when in good order can work perfectly free from any knock, and thus run silently at very

high speeds. The small engines run at over 1,000 revolutions per minute. The one illustrated, which has low pressure cylinders of 20 inches diameter and 9-inch stroke, runs at 350 revolutions per minute or at a piston speed of 525 feet

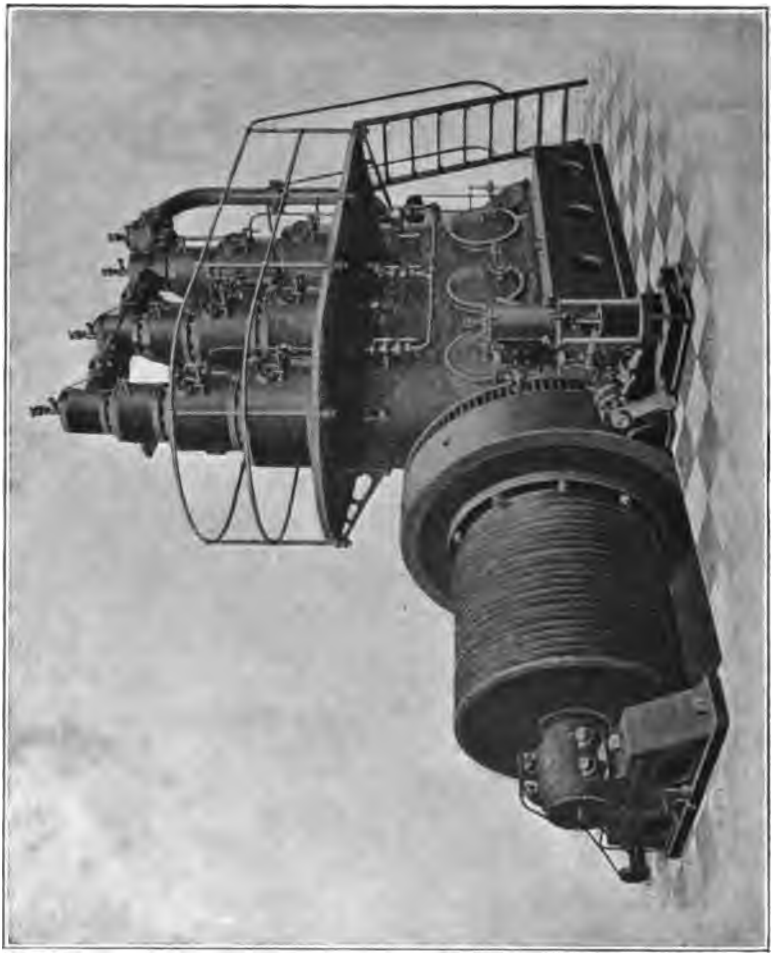


FIG. 262.

per minute. Such an engine will develop 360 H.P., and if worked with a good condenser and superheated steam will require but little over 11 lb. of steam per indicated H.P. per hour.

Though principally used in connexion with electric generators

this form of engine has been successfully used for driving mills and other purposes. In towns and cities where space is valuable and scarce it certainly presents many advantages.

The illustration (Fig. 262) shows a triple-expansion mill engine of 600 H.P. suitable for driving a cotton mill. The driving drum with its twenty grooves for cotton ropes looks ridiculously small in diameter compared with the 30 feet which is sometimes used, but owing to the high number of revolutions it maintains the same speed of driving ropes.

Brotherhood Engine. Another form of quick revolution engine is that known as the "Brotherhood," from the name of its manufacturer, who introduced it about the year 1873. This engine was one of the earliest of these high speed engines, and became widely used, though only for a limited number of purposes. In this engine there are three cylinders equidistant from each other, and arranged around a central chamber as shown in the outline drawing (Fig. 263). Within each cylinder there is a trunk piston, in the outer end of which is fitted the end of the connecting-rod, and there is no piston-rod proper, the trunk serving as such as well as guide. The trunks are bored out to take the ends of the connecting-rod, which have large curved surfaces fitting in what would be otherwise the open ends of the trunks; all the rods work on a central balanced crank. There are three piston valves, one at the side of each cylinder, and all the valves and rods are driven by one excentric within the chamber. These valves serve for the admission of steam only, into each cylinder in turn. There is an exhaust port on the curved end of each connecting-rod and as this oscillates, when the crank end of the rod nears the end of the stroke, a corresponding port is opened in the trunk just for sufficient time to discharge the used steam, but almost immediately closes again, thus imprisoning the remainder of the exhaust, which is compressed on the return stroke. This keeps the connecting-rod brasses always pressed against the crank. This engine will run quietly at very great speeds, and though

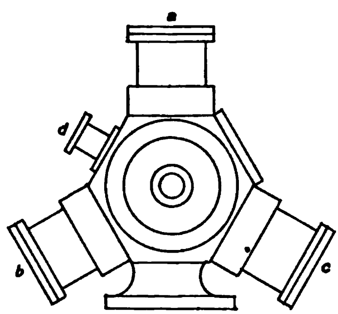


FIG. 263.

not the most economical is very reliable and compact. They are extensively used in connexion with torpedo boats and are then generally worked by compresssed air.

Forced Lubrication. Of late years forced lubrication has been extensively used in connexion with quick-running engines; this consists in forcing oil under a constant pressure of about 15 lb. into all the bearings, introducing it between the fixed and running wearing surfaces. It is contended, and quite reasonably so, that if a film of oil is always kept flowing between the wearing surfaces these latter can never touch each other and thus never wear, the oil alone taking all the pressure. Within very wide limits this is found to be true. The method of working is as follows. A well for the oil is formed in engine base; into this well all the oil is drained, passing through a fine wire strainer on its way. A small force-pump worked from one of the excentrics draws from this well and delivers the oil under pressure to all the working parts by a system of suitably arranged pipes. Some makers drill a small hole right up the crank shaft as far as the crank, up one of the arms, and then to the crank-pin; the ends of the holes are all stopped up with the exception of the entrance for the oil at the end and its exit in the centre of the length of the crank-pin. A channel or tube from one end of the connecting-rod conveys the oil to the other end. In this method some of the oil has to be driven upwards. The main bearings are connected with pipes direct from the oil pump, and the flooded bearings all leak into the oil well.

Another method of oiling the connecting-rod and one which is preferred by the author is to make crosshead slippers of a length greater than the length of the stroke, and make a semi-circular groove in it equal to the length of the stroke. The oil is then pumped through the fixed guide into this groove, filling it constantly with oil at high pressure; a communication is made between this groove and one end of the crosshead pin, and through this, oil is delivered to the small end of the connecting-rod and from there by a tube to the large end, gravity in this case assisting the flow; this method also ensures plentiful lubrication of the piston-rod guide. The main and other bearings are oiled direct from the main pipe as in the former case.

With either method and a good pressure of oil the working

parts are thoroughly flooded with oil, which, after serving its purpose, squirts from all the bearings in fine streams which are all collected in the well for re-use. It is very essential with this engine that the casings all fit oil-tight so as to prevent any loss and also keep the engine quite clean. Apart from the reduction in wear and tear with forced lubrication, there is also the advantage that such engines work much quieter

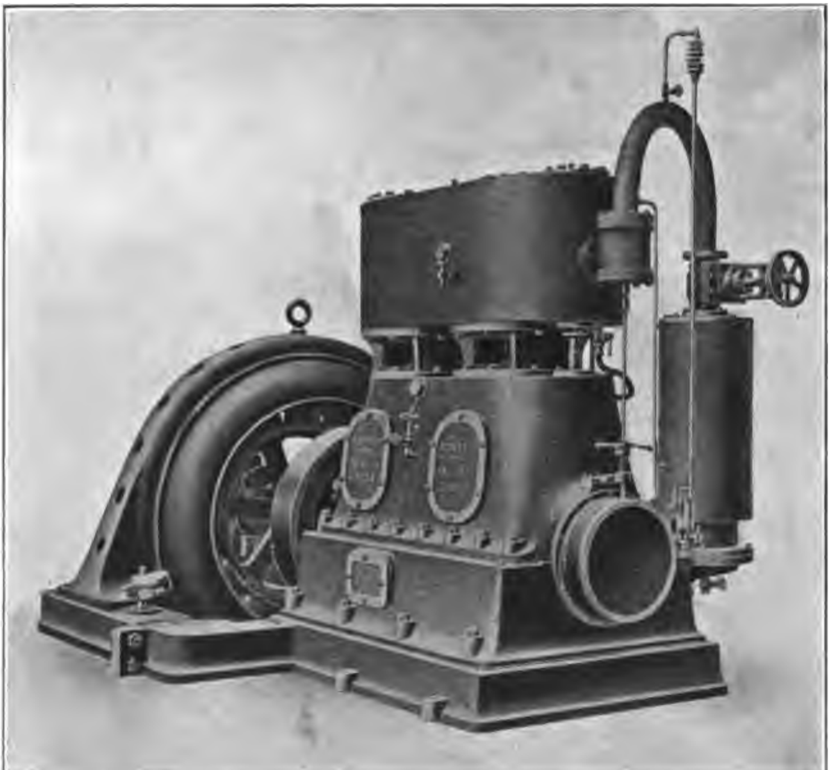


Fig. 264.

and with an almost total absence of knock and vibration, always assuming that the parts are perfectly balanced. A good example of such an enclosed forced-lubrication engine is shown in Fig. 264.

The only drawback to high-pressure lubrication is the difficulty in constantly maintaining the full pressure required, and some makers and users prefer merely flooding the bearings

with oil at atmospheric pressure. The oil well and pump are used, but the latter is generally a small centrifugal or rotary pump, delivering a large quantity of oil at low pressure. There is not any squirting of the oil in such cases but almost as much splashing with quick-running engines, and any splashing of oil away from the engine is wasteful, and in the case of direct-driving electric generators is very prejudicial, hence

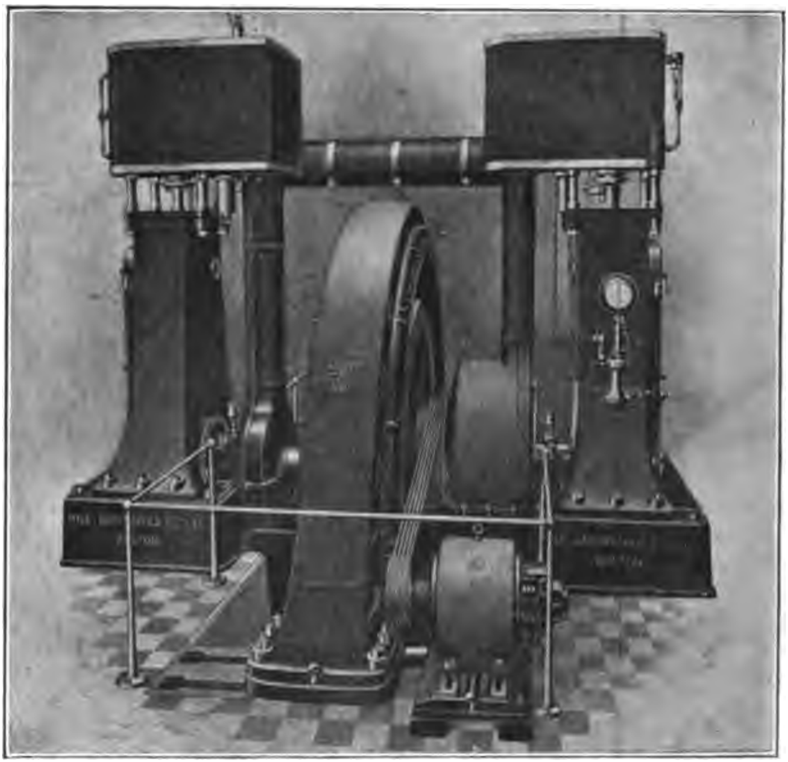


Fig. 265.

such engines are equally well enclosed within casings (Figs. 265 and 266).

The former is a slide valve vertical compound engine by Hick, Hargreaves & Co., with an alternator between the two standards and an exciter driven by ropes from a pulley on the shaft. The high pressure cylinder has a piston valve, and the speed is regulated by a shaft governor within the

large casing. On the high pressure side all the working parts are enclosed within the frame with the exception of the excentric and rods, which it will be seen have separate casings of their own. Such an engine can be flooded with oil inside and still be perfectly clean outside the casing.

Fig. 266 shows a large vertical engine of 500 H.P. by Robey

Fig. 266.

& Co. This is a comparatively quick speed engine having a 2-feet-six stroke and making 150 revolutions per minute. It is fitted with Corliss valves, but has no wrist-plate or releasing motion to the valves, the method of working which is clearly seen in the illustration. The engine delivers its power at the side instead of in the centre and is suitable for driving shafting direct from one end, as is often required; there being no releasing gear the engine works very quietly, and being

intended for a constant regular load the cut-off can be fixed at the most economical point in both cylinders and the governing done by the throttle valve. It will be noticed that the starting wheels are so arranged that the engine can be stopped and started either from the platform or the ground with equal facility.

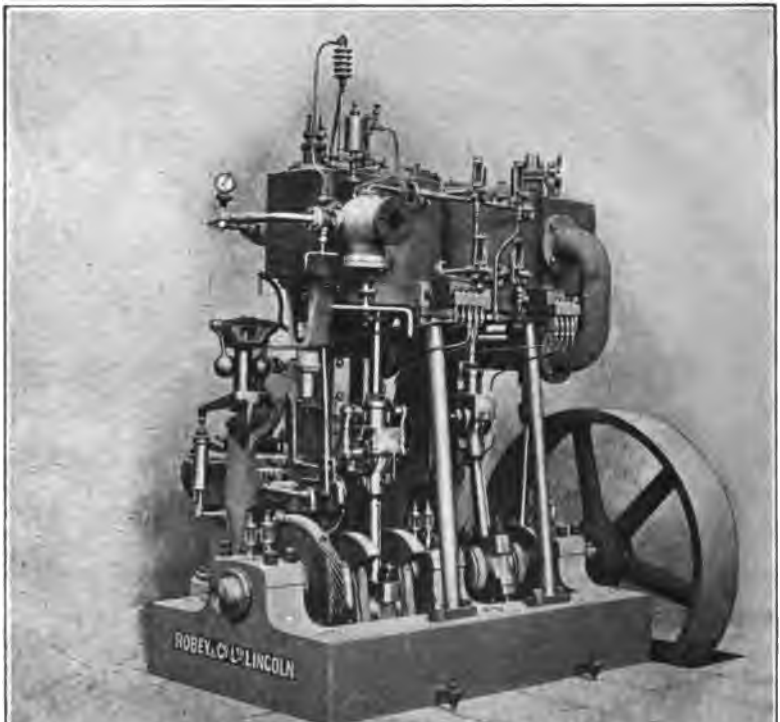


FIG. 267.

In spite of the many advantages of forced and splash lubrication there are still many users of engines who prefer to have all the working parts of the engine clearly visible and easily accessible. Fig. 267 is a good illustration of an open-fronted compound vertical engine fitted with link expansion gear controlled by a Richardson governor. This engine has a stroke of 12 inches and can run up to 300 revolutions per minute.

Behind the high pressure standard there is an air pump and condenser. The air pump is worked from a lever which reduces the stroke 4 to 1, so that the air pump has only a 3-inch stroke and works quite easily and quietly at the high speed above mentioned.

Drop Lubrication. The lubrication of all the parts at the high speed is not very easy, but is satisfactorily done from two comparatively large oil chambers fixed one in front of each cylinder. From this chamber pipes with separate adjustment to each are led, one to each wearing surface which is in motion; these pipes deliver the oil drop by drop as it is required. The crank-pin, which is the most inacessible, has a tube reaching from the large end brasses of the connecting-rod up to the upper end, where it terminates in a small cup containing cotton-wool. As the connecting-rod comes to the top of its stroke the wool in the cup just touches the end of the oil pipe and occasionally takes from it a drop of oil, which it conveys to the crank-pin. The same thing is done with the excentrics and other moving parts, the main bearings having sight-feed lubricators of their own. With this method of lubrication there is a tendency to sprinkle some oil from the connecting-rod, but this is caught and conveyed to the oil well by a piece of sheet steel in front of each crank.

Tandem Horizontals. Before leaving this type of high and medium speed engine we illustrate a medium stroke compound horizontal engine with the cylinders placed tandem to each other; this is shown in Fig. 268. All the parts of this engine are exceedingly strong and substantial, and all the working bearings are of large size. When there is length to spare and but little width a compound engine of this type makes a very good prime mover. The way in which the two cylinders are joined together is worthy of notice. The conical casting shown is divided into two parts longitudinally, and the parts are securely and truly bolted together by the longitudinal flanges; it is then accurately bored out at each end to fit on a turned surface on each end of cylinder outside the covers. When all is bolted together as shown, the two cylinders are centred and held in perfect line with each other as they should be, but when the cylinder cover requires to be taken off, the distance piece can be easily removed by dividing it again into its two parts.

Sometimes such tandem engines are placed side by side with a large driving wheel between them, the two engines being controlled by one governor. The principal advantage

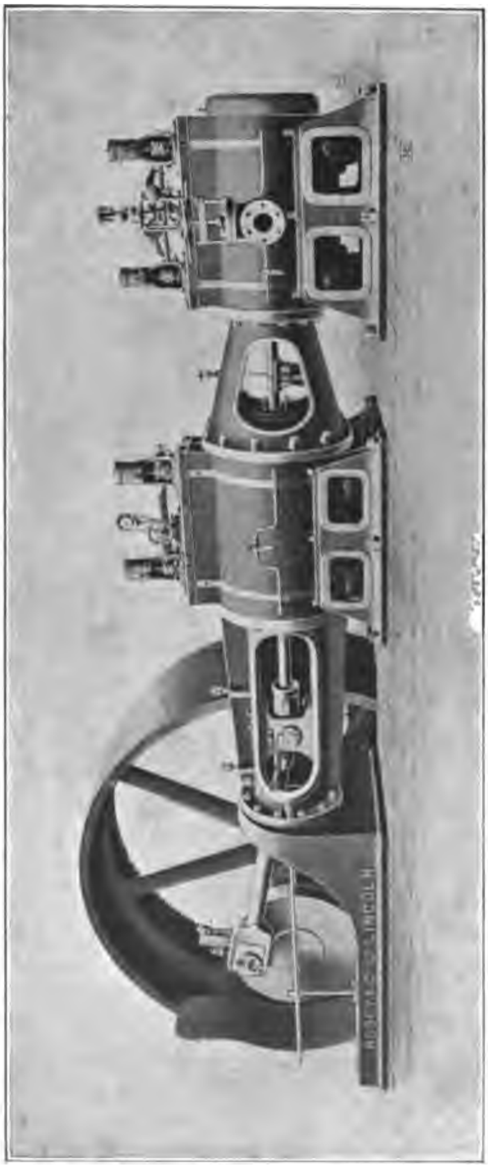
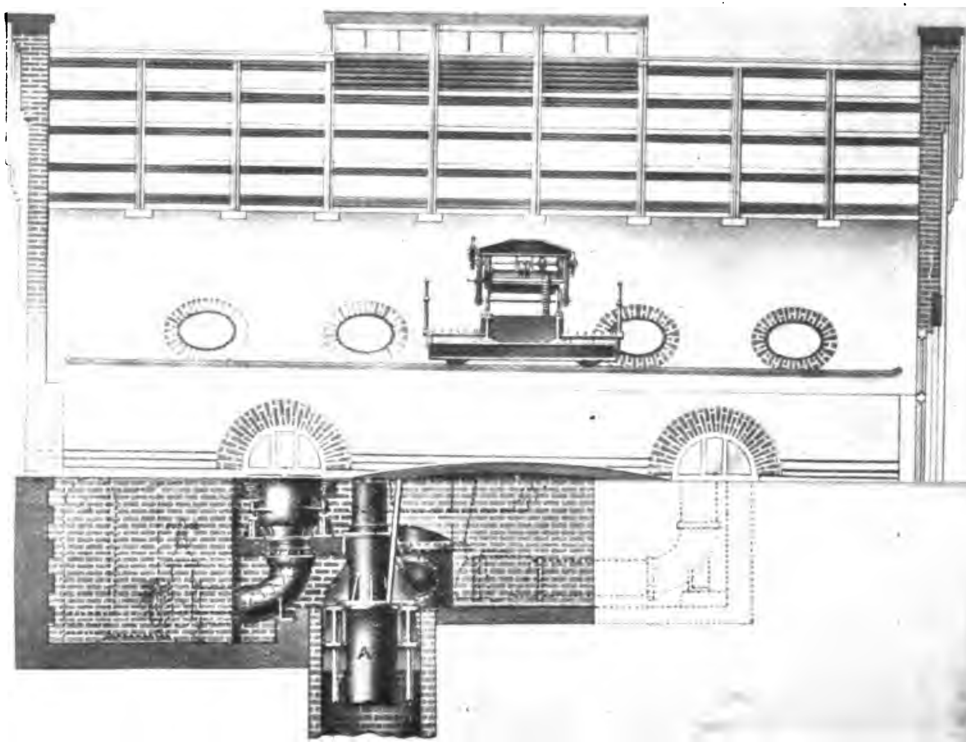


Fig. 268.

[Plate III.



Beam Pumping Engine for Chatham and Rochester Water-Works, by Simpson & Co., London.

[To face p. 311.

of this arrangement is to get a more equal and continuous pressure on each of the crank-pins at all points of the cut-off, and when there is extreme and frequent variations in the power this course is advisable.

Beam Pumping Engine. The beam engine, though one of the earliest forms in which steam power was utilized, is now but very seldom used, owing principally to its much greater cost in comparison with other types, and to the fact that for most purposes there is no advantage to compensate for the extra cost. In the early days of the steam engine, the raising of water was the principal use to which steam power was applied, and after more than a century of development of the various types the beam engine is found to be still one of the best and in the opinion of some experts quite the best form of engine for this purpose.

We give, therefore, a large illustration (Plate 3) of one of these engines of the most modern construction by the well-known firm of Simpson & Co., who have had probably the largest experience in pumping engines of any other firm, and the drawing which they have kindly prepared for this work will be found well worthy of study.

This engine is a compound rotative surface-condensing beam engine, with high-lift and well pumps. It was constructed for the Chatham and Rochester waterworks, and is designed to raise 3½ million gallons per twenty-four hours from the bottom of a deep well, and deliver it into a high service reservoir, the combined lift being 380 feet.

Pumps. The two sets of pumps are worked direct from the beam, the well pump being the one nearer to the centre. Both sets are bucket and plunger pumps. The pump *A* draws its water from the bottom of the well and delivers it into a tank at ground level. The water enters the surface pump *B* through the pipe *C*, a valve at *D*, shown in dotted lines in the suction chamber, enabling the pipe *B* to be cut off when required. The water is delivered to the high-level reservoir by the pipe *D*, the large air vessel *E* serving to equalize the pressure in the delivery pipe and to prevent shock.

There are two cylinders, one high and one low pressure, both connected to the beam by the same parallel motion. The high pressure cylinder being nearer to the centre of beam has in consequence a shorter stroke, and it is shown placed

at a higher level. It receives its steam direct from the boiler and exhausts into a receiver below it, which receiver is fitted with steam tubes under boiler pressure and acts as a re-heater. From the receiver the low pressure cylinder draws its steam and discharges the same by the pipe *F* into the surface condenser *G*, the delivery water from the well pump *A* passing through the condenser by the pipe *H* on its way to the surface-level tank. The air pump, worked direct from the beam, is shown at *J*; the pump delivers the condensed steam in the form of hot water to the hot well, from which the feed pump *K* returns it to the boiler.

Steam Distribution. The steam distribution valves and gear are of the Corliss type, and are driven as follows. A bevel wheel on the crank shaft drives a countershaft *M M* parallel with the length of beam; another pair of bevel wheels of equal proportions *L* drives the excentric shaft situated below and between the two cylinders; on this shaft, revolving in equal turns with the crank shaft, there are four excentrics, two for the admission and two for the exhaust of the two cylinders. The valves on both cylinders are actuated independently from the wrist-plate in the centre of the side of each cylinder. The admission valves of the high pressure cylinder are fitted with tripping cut-off gear, and are controlled by a very sensitive high speed governor; those on the low pressure cylinders are adjustable by hand while the engine is running.

Speed Adjustment. The speed of the engine is capable of ready variation to the greatest nicety by means of a wheel adjustment on the governor while the engine is at work, and there is also a safety device incorporated with the governor, whereby, should the speed of rotation vary within narrow limits above or below the predetermined normal, the high pressure valves are thrown out of action and left in a position covering the steam inlet ports, whilst at the same time a valve on the condenser is opened and the vacuum destroyed.

"Duty" of Pumping Engine. This engine will develop 300 H.P. in water lifted, with 100 lb. steam pressure in the boiler, and will maintain a duty of 125,000,000 in normal running with Welsh coal. The "duty" of a pumping engine is an expression which has come from Cornwall, the birthplace of this type of engine, and it is stated as 60,000,000 lb. of water raised 1 foot high by one bushel of Welsh coal (a bushel

weighs 94 lb.). Like many of the old engineering standards, it is complicated and imperfect, but it is still frequently used when dealing with the raising of water. To bring it into modern phraseology and reduce it to pounds of coal per horse-power per hour, we must divide the number 166·32 by the duty in millions, the result is pounds per horse-power per hour; thus, with a duty of 125 millions' $\frac{166 \cdot 32}{125} = 1 \cdot 33$ lb. of coal per horse-power per hour, an exceedingly good duty for a pumping engine.

One pound of Welsh coal is able to evaporate 10 lb. of water, so measured in steam this equals 13·3 lb. of steam per horse-power per hour, a very satisfactory result when we consider the low pressure of steam (100 lb.), and the necessarily low speed of piston.

When in order to make comparison we want to change pounds of coal per horse-power per hour to "duty," we have to multiply 166·32 by 1,000,000 and divide the result by the pounds per horse-power per hour, this then being 1·33 lb.

$$\frac{166 \cdot 32 \times 1{,}000{,}000}{1 \cdot 33} = 125{,}000{,}000.$$

While such an engine as we have described above is probably the best which can be used where, as in this case, the water has to be drawn from a deep well and forced to a great height, yet when the principal work is forcing the water, there are other forms of pumping engines which have special advantages of their own. Amongst these what is known as the "Worthington" pumping engine will be next described.

Worthington Pumping Engine. A drawing of the side elevation of the latest form of this type of engine is shown in Fig. 269 and in section in Fig. 271. These engines are always supplied in pairs as shown in the outline plan (Fig. 270). Part of the valve gear of one set of engines being worked from the crosshead of the other set makes this necessary, and there is the further advantage that as one pump ram is at the end of its stroke that of the sister engine is moving at its most rapid rate, so with the two sets there is a quite continuous discharge of water at a practically uniform velocity.

Coolgardie Water Supply. The illustrations given are taken from the working drawings of the engines supplied to Coolgardie in Western Australia, one of the largest and most

important waterworks installations in the world. Coolgardie was practically destitute of fresh water, and the need being

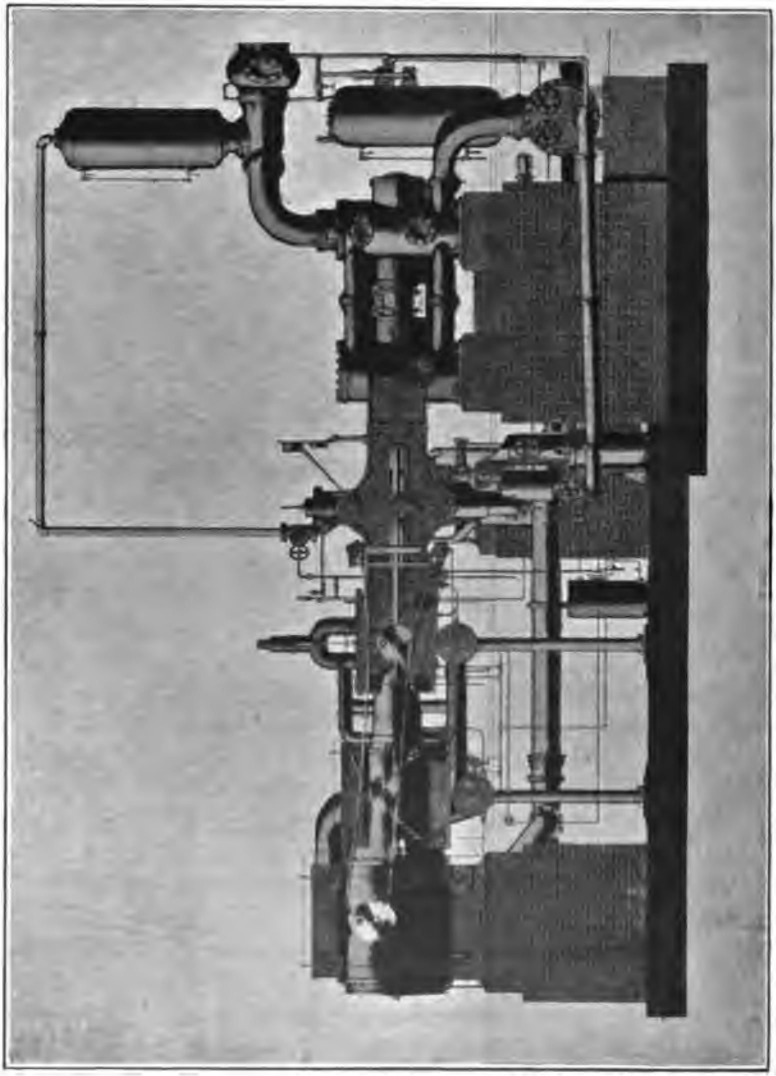


Fig. 269.

great it was brought a distance of 363 miles from the source of supply and over a range of granite ridges averaging 1,200 feet high. The total distance was divided into eight sections,

and twenty pairs of pumping engines were installed, delivering to Coolgardie 5,600,000 gallons of water per twenty-four hours. There are twenty pairs of engines. Each pair of engines has two high pressure cylinders 16 inches diameter, two intermediate cylinders 25 inches diameter, and two low pressure cylinders 46 inches diameter. Twelve of them have double-acting rams 15 inches diameter, eight of them have double-acting rams 21 inches diameter, and all have a stroke of 36 inches in length. The whole of the steam cylinders are jacketed with steam at boiler pressure, viz.. 175 lb. per square inch, the cylinder covers being jacketed as well. The estimated I.H.P. of each duplex engine may be taken at 300.

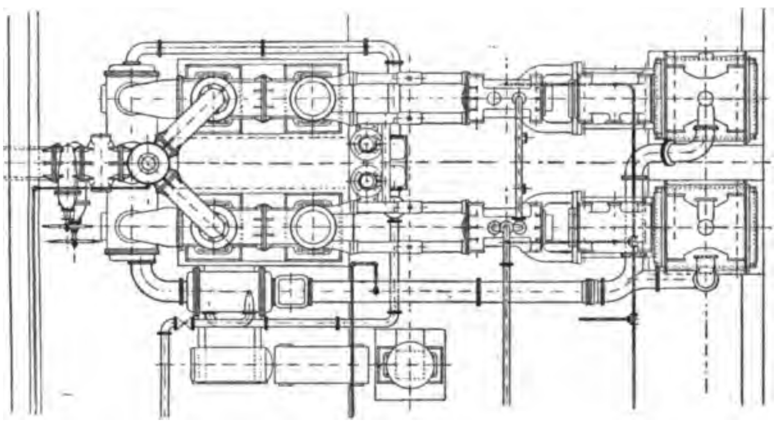


Fig. 270.

The steam valves are semi-rotative of the Corliss type, and are driven by the Worthington patent high duty valve gear. Intermediate re-heaters with steam at boiler pressure are used between the high pressure and intermediate cylinders, and also between the intermediate and low pressure cylinders and, as is clearly shown in the drawing, below the high pressure and intermediate cylinders respectively.

The construction of the engines can be best understood by referring to the longitudinal section (Fig. 271); from this it will be seen that the low pressure piston and the intermediate pistons are connected together by the piston-rod common to both, a metallic stuffing-box being between the two in the

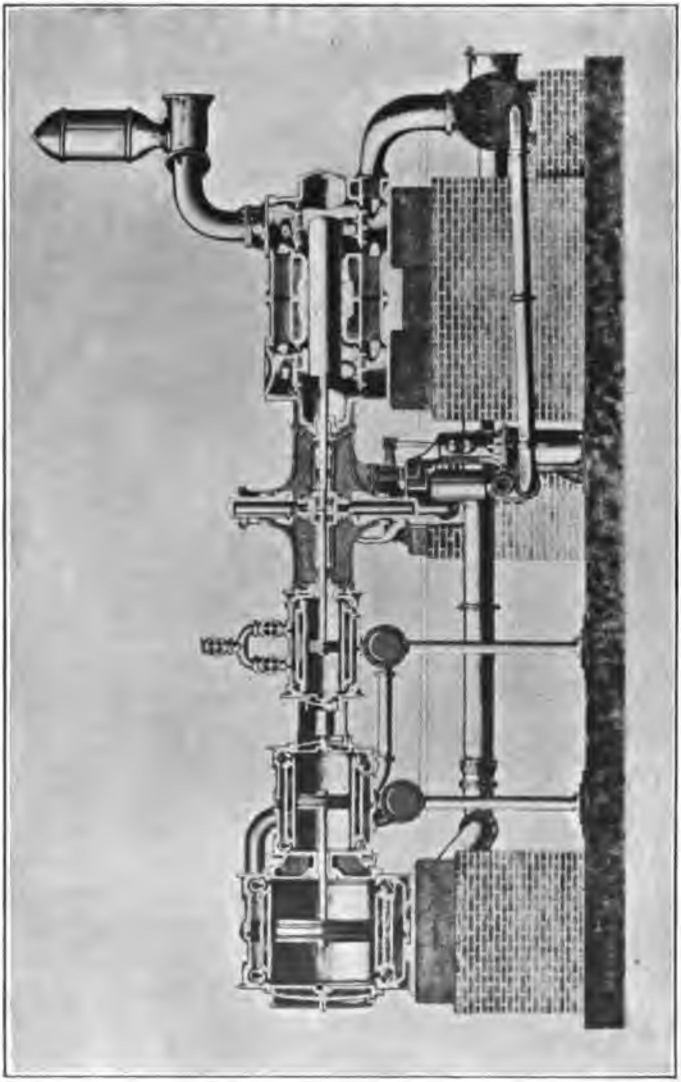


Fig. 271.

cover which is common to the two cylinders. The high pressure piston is connected direct to the pump rams through the crosshead, which works in guides in the frame. This crosshead extends outside the frame on each side, and rods from it extend backwards past the high pressure and intermediate cylinders, passing through stuffing boxes into the low pressure cylinder, and being attached to the low pressure piston, one on each side, there being three piston rods to the low pressure piston. By this arrangement all the pistons are connected together and in a better way than by putting all of them on one rod. A reference to the sectional drawings (Fig. 271) will show that one cover of each cylinder can be removed and the piston examined or repaired without disturbing any of the others. The covers as well as the cylinders are all steam jacketed and well lagged; this is very important owing to the comparatively slow speed at which the piston works.

The path of the steam is as follows: Entering at the stop valve above high pressure cylinder (Fig. 271) it goes into the branch pipe; to each leg of this pipe there is a regulating valve, by means of which the forward and backward strokes of the rams can be set to exactly the same speed. The Corliss valves at each end admit the steam to the cylinder, and it emerges through the two Corliss exhaust valves into the re-heater fixed below the cylinder, thence by pipes coming out at the left to the steam chamber above the intermediate cylinder, in which chamber the admission valves for this cylinder work. It emerges through the exhaust valve at the bottom of the cylinder into the other re-heater fixed below it. From thence it comes into the valve chamber above the low pressure cylinder. Emerging from the exhaust valves at the bottom it passes through a Webster oil separator and hence to an auxiliary feed-water heater of the tubular type, thence through the surface condenser at the extreme right of the drawing; it is then condensed into water, and emerging by the smaller horizontal pipe, it enters the air pump, which is fixed under the guide frame and is delivered into the hot well.

An ingenious arrangement for using the jacket steam has been carried out; the steam is used to work the feed pumps, which are of the usual duplex type, and they exhaust into the

"Webster" feed-water heater. By this means the feed water is sent forward from the water heater through the "Greens" economizer at a very high temperature.

In direct-acting pumps like these the pressure against the pump rams is equal throughout the stroke, while the pressure on the steam pistons is a rapidly diminishing one, when, as in this case, for the sake of economy the engine works with a high rate of expansion. The result of such an arrangement would be an unnecessary and dangerous velocity at the commencement of the stroke, slowing down and possibly coming to a stop before the end of the stroke.

The way in which this difficulty in the way of high expansion is overcome is exceedingly ingenious and effective. In the drawing (Fig. 271) the crosshead is shown in the middle of its stroke, and there are two pressure rams, one above and one below the crosshead guide, both connected with the crosshead by ball and socket coupling. The casings of these rams are pivoted on trunnions in the engine guide frame, clearly shown in Fig. 269. How they act to convert the varying pressure of the piston to a constant pressure on the rams is best seen by referring to the outline diagrams (Figs. 272, 273 and 274).

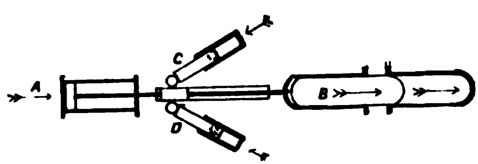


Fig. 272.

In Fig. 272 the pistons are just starting on their outward stroke with the highest pressure upon them; the arrow A shows the direction. The ram is meeting with only its normal resistance, but the two supplementary rams are also opposing the movement of the pistons, and the result of the combined opposition is that the piston and pump ram start steadily at the regulated speed. As the pressure on the steam piston falls, so does the resistance of the rams, C, D, till when the piston and pump ram have made half the stroke the resistance of rams C, D vanishes altogether, they having now assumed a vertical position where the resistances being equal and opposite, exert

no force in the direction of travel. This is shown in Fig. 273.

The moment, however, that the piston has passed the centre line of the travel, the rams *C*, *D* begin to exert a pressure in the direction of travel instead of a resistance to it; this pressure increases as the angle of the rams with a vertical line

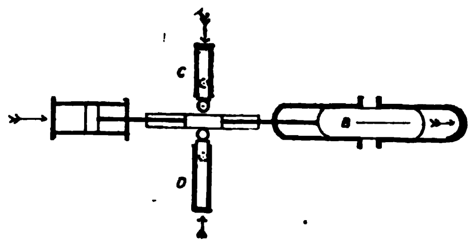


FIG. 273.

becomes greater; thus when the steam pressure is least on the piston, the ram pressure helping them forward is greatest, as in the position shown in Fig. 274. The inertia of the moving parts then, in their resistance to being moved from a state of rest at each end of the stroke, and their tendency to persist in the movement given to them, also helps in this direction, the sum of all these forces being so regulated that a regular and uniformly increasing and decreasing motion is obtained from one end of the stroke to the other, and the delivery of water from the combined pair is perfectly continuous and uniform.

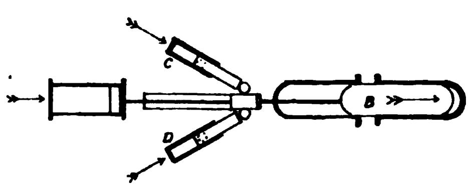


FIG. 274.

The expansion gear can be adjusted to run at speeds varying from 30 to 180 feet of piston per minute, and will continue at the speed fixed so long as the conditions remain the same. The compensating cylinders are connected through one of the trunnions to the water in the rising main, and are thus

exposed to the same pressure; thus should an accident occur and the main be burst the pressure on the cylinders would be at once removed, and deprived of this help, the pump ram could not finish the stroke; the engine would therefore at once come to a standstill.

The method of working the Corliss valve is very ingenious, and can best be seen by reference to the large drawing in Plate IV. The engine there shown is similar to the one above described, only it has a jet condenser and an air pump worked by a rod from the pump ram, instead of the surface condenser shown in Fig. 270. The valve gears, however, are identical.

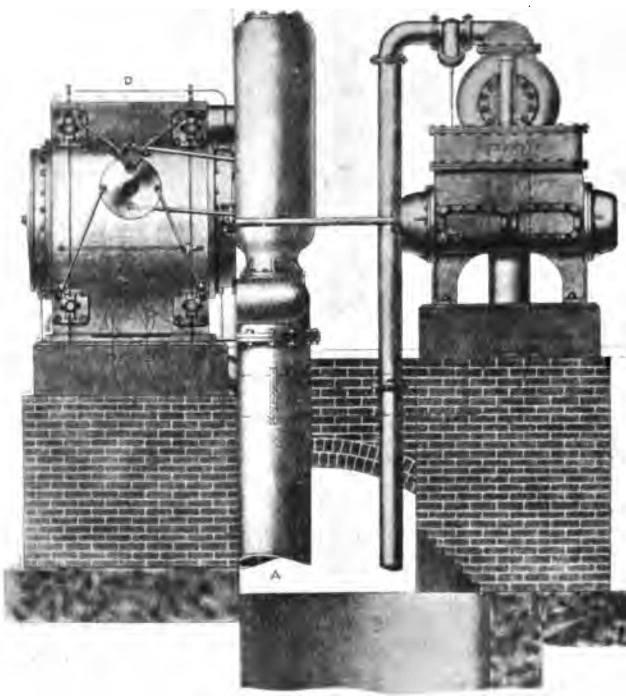
Here a strong bell crank lever *A* is worked from the crosshead; by means of the vertical rod shown it gives a reciprocating motion to the shaft *B* with its levers. From this shaft the wrist-plate of the *sister* engine is worked, and correspondingly the shaft *C* with its lever is actuated by a similar lever worked from the crosshead of the sister engine.

The crosshead of the sister engine is moving at its most rapid rate in the middle of its stroke when this engine is at the end of its stroke, hence the valve can have a rapid opening and closing motion. It is the exhaust valve alone which are worked direct from the wrist-plate by the lower set of rods and levers.

The admission valves are worked through the upper set of rods and levers direct from the lever *A* of its own engine. The wrist-plate for the admission valves is pivoted on a point near the circumference of the exhaust wrist-plate; this arrangement gives a rapid opening to the admission valve though the lever working them is at that moment quite at rest. Means are provided for fixing the cut-off at any desired point, and the result of all is a high rate of economy equal to, if not surpassing, that of any other form of pumping engine, and greatly above the average of such engines; while, besides, this form of direct acting engine has many advantages peculiar to itself.

Besides the many different engines referred to and described in the former part of the book, we have in this chapter dealt as fully as space permits with fifteen different types of steam engines. The variety in design is endless, and it is manifestly impossible to deal with all. Those selected for illustration are fairly representative of what is being done by the most

[Plate IV.



[To face p. 320.

celebrated makers in the various lines. Without doubt there are many others entitled to stand on the same level, but those given are examples of the best and most modern practice both in design and construction, and will be found well worthy of the most careful study.

The author takes the opportunity of again thanking the many engineers who have kindly prepared drawings and illustrations of their manufactures for this work.

CHAPTER XVI

Feed and Steam Heating

WATER heaters and superheaters are on the border line between the boiler and the steam engine. The former pertains more to the boiler; we shall deal with it only shortly.

It must be evident that the hotter feed water can be made before entering a boiler, the smaller quantity of additional heat will be required to make the water into steam—hence the value of a water heater.

These heaters are made of two kinds, those which heat the feed water by the exhaust steam, and those which heat it by the waste heat in the flues of the boiler.

There are two ways of heating by the exhaust steam, i.e. by direct contact, or by the surface contact of tubes containing it.

When, as is often the case, the feed water contains a large amount of solid matter, this should be removed before it enters the boiler; some matter may be removed by merely settling or precipitation, and when this can be done, settling tanks should be provided. Other matter can be removed by filtration, and this can be provided in connexion with the settling tanks. It very frequently happens, however, that solid matters are dissolved in the water, such as many salts of minerals, often as sulphate and carbonate of lime, and these are precipitated only at a high temperature. A direct contact heater is useful in such a case, and many such are made. They consist generally of a vertical chamber of large capacity (i.e. about six times the area of the steam cylinder), and of a height about four times the width. The chamber is fitted with rectangular shelves, as shown in the outline drawings (Fig. 275). The exhaust steam enters by the pipe at A, passing backward

Direct Contact Water Heater.

and forward over the plates, and emerges in a much diminished volume at B. The feed water enters by a distributing pipe at C and falls in a series of drops on to the first plate; these plates are better corrugated so as to keep the water in tiny thin channels extending over the whole width, as without this provision there would be a tendency for the water to get into a narrower stream and thus expose a smaller surface to the steam. Flowing, as it does in this heater, backwards and forwards over a large surface, exposed to the action of hotter steam as it nears the bottom, the water attains a high temperature and parts with the soluble salts, depositing them upon the plates, which are taken out and cleaned periodically. The efficiency of the heater suffers but little, if anything, from the deposits, as the heating is all done on the surface. The steam meeting the cooler water rapidly parts with its heat, and a good deal of it is converted into pure hot water, joining the feed. The connexion of the heater to the feed pump is at the pipe D. There is one drawback to these advantages. This heater needs either to be fed from a natural or town supply at the height of the distributing pipe C; or the engine must have two feed pumps, one drawing from the well or other supply below the heater, and the second to pump from the heater into the boiler.

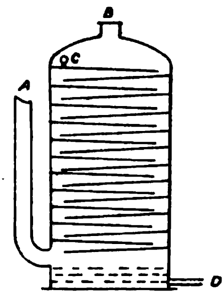


FIG. 275.

Feed-water Heater. The second form of steam water heater is practically a small surface condenser, and consists of a nest of tubes around which the exhaust steam is passed, the feed water going within the tubes from the pump on its way to the boiler; the tubes are generally about 2 inches diameter, and facilities are provided for cleaning their internal surface if they get coated with incrustation from the water. Water in this way can be raised about 100° in temperature, but the more perfect is the expansion of the steam the lower will be its temperature of exhaust, and the less efficient will be the water heater.

Such heaters may be, and often are, placed between the cylinder and the condenser of a condensing engine with very good results.

Boiler Flue Heater. A more effective way of heating the water, and one by which a much higher temperature is obtained, is by the boiler flue heater. If we are working with steam of 200 lb. pressure above atmosphere, it will have a temperature of nearly 400°,[1] while the gases escaping from the flue into the chimney may be from 500° to 600° temperature. There is so little difference between these two temperatures that, as was shown in chapter IV, the heat passes from the one to the other at a much slower rate than if the difference in temperature was greater ; and any further extension of the heating surface of the boiler is practically without result. If now we introduce into the flue a low temperature boiler with a large heating surface, and pump our feed water through it, we shall get a much better result. We start with water with a temperature of say 60°, and introduce it into gases at 600° ; we have then a difference of temperature of $600-60$, or 540° ; the transfer of heat will, therefore, be much more rapid, while the temperature of the flue gases will be but little lowered, because the volume is so much greater.

This form of heater generally consists of a large number of tubes about 4 inches diameter, arranged in vertical rows in a flue chamber built for the purpose between the boiler and the chimney, and occupying a space often half as much as the boiler itself. The tubes are of a length about equal to the diameter of the boiler, and joining at each end to a transverse tube of larger dimensions. The lower of these transverse tubes is of much larger diameter, and serves to collect the scale from the inner surface of the tubes and the deposit from the water. This deposit is taken out from the ends, which project through the brickwork on the outside. Each vertical tube is surrounded by suitable scrapers which are moved mechanically up and down at a slow rate, and thus keep the surfaces free from soot and flue dust, which would otherwise prevent the free transmission of heat. Though these are now made by several manufacturers, yet from the name of the original patentee they are usually known as Green's Economizers.

Superheater. Another source of economy in the use of steam consists in heating the steam after it has left the boiler to a higher temperature than that of the steam in the

[1] 387·6°.

boiler; this is called superheating; the advantage of doing this was recognized as far back as the middle of the last century. At that time 60 lb. was considered a high pressure, and the economy arising from superheating such steam was very considerable, so much so that it was largely adopted, especially in marine practice.

It is well known that when steam is heated in contact with water, i.e. in the steam space of a boiler, its pressure increases with its temperature, and more of the water is evaporated.

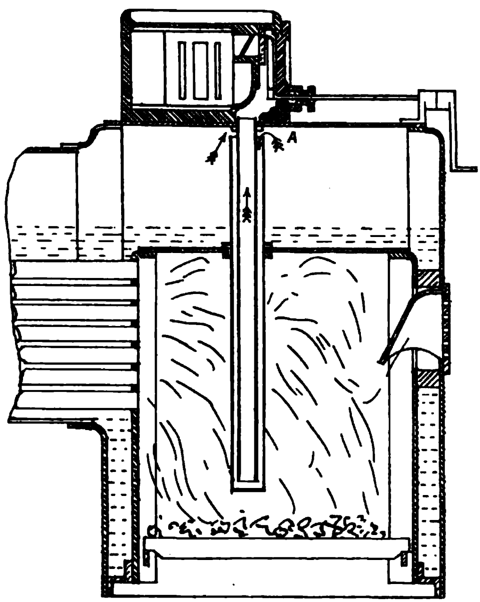


FIG. 276.

When, however, steam is heated after it has left the boiler and is on its way to the engine, its pressure remains the same, but its volume increases; this extra volume passes into the steam engine cylinder and thus reduces the quantity of steam taken from the boiler.

In the early days of superheating, when only animal and vegetable lubricants were used, the rapid deteriorations of valve faces and other surfaces exposed to friction under steam pressure was so great as speedily led to the abandonment of

superheating ; and when the heaters were used in many steamships, it was found necessary to inject cold water into the steam supply in order to keep working at all. Further, the adoption of much higher pressures, the introduction of surface condensers and compound engines, provided sources of economy more satisfactory than superheating the steam.

Among the many early superheaters was one patented by the author in 1865, and shown in Fig. 276. This was for use with portable engines. The illustration shows a section through a fire-box and boiler shell and steam chest. The inner tube shown was screwed into the crown of the boiler, and was open at each end. At the upper end it communicated with the steam passage into the steam chest.

The tube was surrounded by a larger one, open at the top and welded up at the bottom ; the large diameter tube came within about 9 inches of the fire-bars, and was exposed to all the strongest heat of the fire, most of the flame having to pass it on the way to the boiler tubes. The steam entered where shown by the arrows at *A*, and, descending the annular space between the two tubes, entered the smaller tube at the bottom, and thence to the steam chest. In doing this the steam was raised to a high temperature, and the result was a very considerable economy in fuel. The same difficulty that has been mentioned above arose with the valve faces, and the use of the superheater was abandoned ; it was, nevertheless, a simple and effective apparatus and might at the present time be used with advantage.

With the approved lubricants and metallic packings which are now available, superheating has again come to the front ; and, during the last few years, has been very largely used ; this time principally in connexion with land boilers and engines.

Among the principal superheaters now in use is the one manufactured by Messrs. Simpson & Sons, the makers of the pumping engine shown in Fig. 269, and illustrated by Fig. 277, which shows only the end view of a horizontal boiler. These heaters are made entirely of steel ; they cannot be fixed conveniently in the furnace, but are placed as near to it as convenient, immediately behind the furnace flue, where they are exposed to a temperature of from 1,000° to 1,500°. The steam from the boiler passes by the pipe *A* to *B*, where the pipe is enlarged to take a group of pipes, five in number, and each

of 2 inches diameter, the length and number of the groups being determined by the size of the boiler. In the illustration there are four groups, or twenty pipes in all. These groups, or elements as they are called, are connected together at top and bottom by bent pipes of larger diameter; and at *C* they are gathered together again into one pipe, from whence the steam is conducted to the engine in a highly heated state.

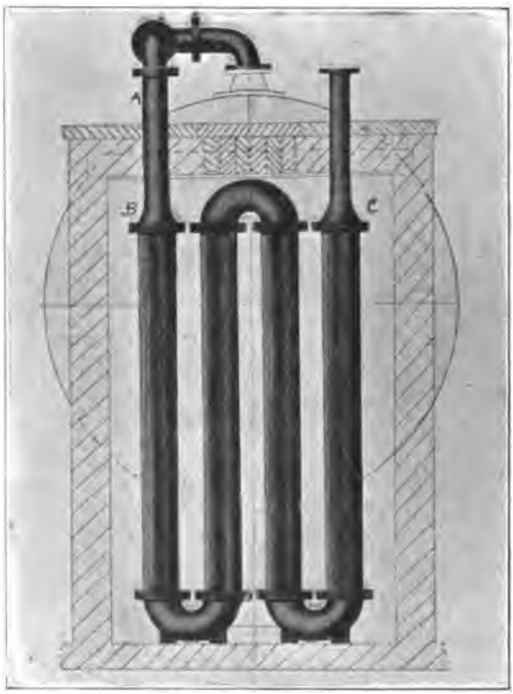


FIG. 277.

Another form of superheater manufactured by Messrs. Hick, Hargreaves & Co., the makers of the Corliss engine shown in Fig. 125 and others later, is shown in side and end elevation in Fig. 278 and 279. These superheaters, like the preceding ones, are fixed at the end of the furnace flue. The apparatus consists, as is seen by the illustrations, of a number of U-shaped tubes of small diameter and great thickness, made of a special grade of steel, being solid drawn without a weld. The tubes

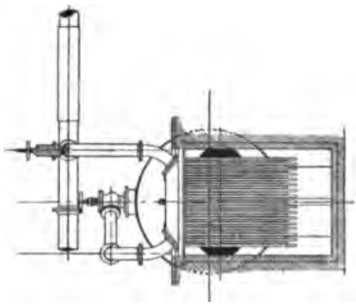


FIG. 279.

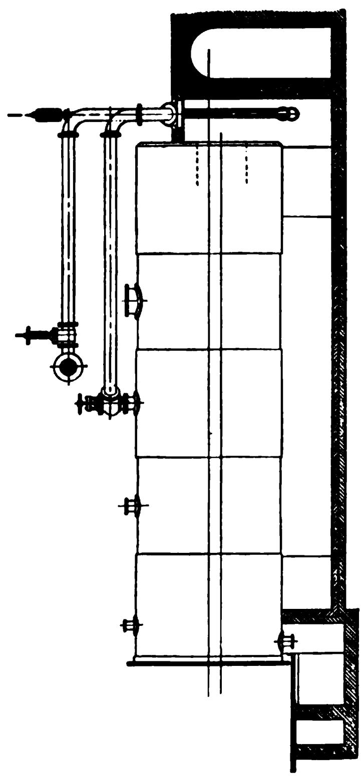


FIG. 278.

hang vertically from a heavy wrought steel tube plate, into which they are expanded like ordinary boiler tubes. The tube plate is heavily bolted to a cast steel box *A*, provided with inlet and outlet branches, and it is divided longitudinally by a diaphragm, so arranged as to cause the steam to pass in parallel down one leg of each tube and up the other. For an 8 feet Lancashire boiler with a steam pipe 7 inches diameter, the hanging tubes are alternately 6 feet and 6 feet 6 inches long and 33 in number.

They have been in use now for over ten years with very satisfactory results.

When engines are fitted with slide valves and have ordinary soft packing, 50° to 60° of superheat is as much as the author would advise, and this is of considerable value. But engines having Corliss or drop valves such as are illustrated in the preceding pages, are eminently suited for working with superheated steam, for which purpose they would naturally be fitted with metallic packing in all the glands. Messrs. Hick Hargreaves claim to have worked engines without any difficulty with steam superheated to 500° and 600°, and, in one instance, as high as 700°, a Continental authority claiming to have used 750°, at which point the engine began to give trouble. These are, however, exceptional cases. The greatest heat the author has used with success is 100° above the temperature of steam at 200 lb. pressure, which is 387·6; this makes a total heat of nearly 500°, beyond which he would not recommend steam to be used for long periods, and at which temperature very satisfactory results can be obtained. Without question, with this degree of superheat and with a triple expansion engine working with steam of 200 lb. pressure, and exhausting into a good condenser, the steam used is not more than 10·5 lb. per horse-power per hour.

Many reliable tests show that superheating to 500° does effect an economy of from 15 per cent. to 30 per cent., according to the class of engine used.

Some carefully conducted experiments made by Mr. C. C. Leach at the Seghill Colliery, give very interesting results, as will be seen by the following table.[1]

[1] *Transactions of the Institute of Mining Engineers*, September, 1902.

Test of Superheating at Seghill Colliery.

No. of Test.	Superheating or not Superheating.	Date.	No. of Boilers at Work.	Description of Coals used.	No. of Days of 10 Hours Coal-work.	Average Weight of Coal burnt per Day.	Ash.	Average Drawings.	Average Per cent. of Coal used to Drawings during Coal-work.	Average Total Indicated H.P., Donkey-Pumps and other small Engines being estimated.	Average Weight of Coal burnt per Indicated H.P. per Hour.	Indicated H.P. per Boiler.	Coals burnt per Sq. Ft. of Fire-grate per Hour.
						Lb.	Per cent.	Tons.	Per cent.	H.P.	Lb.	H.P.	Lb.
1	Not Superheating	Nov., 1901	4	Small coals with about 16 per cent. of rough nuts	8	31,537	12·40	1,577	0·8928	276·4	11·41	69·1	22·63
2	Superheating	Jan., 1902	4	Do.	1	29,664	14·22	1,671	0·7920	266·0	11·15	66·5	21·29
	Saving on	No. 1 test	—	—	—	1,873	—	—	11·29	—	0·26*	—	—
3	Superheating	Jan., Feb., March and June, 1902	3	Do.	8	27,335	11·04	1,691	0·7218	276·2	9·897	92·1	26·16
	Saving on	No. 1 test	1	—	—	4,202	—	—	19·15	—	1·513†	—	—
4	Superheating	June, 1902	2	Nuts . . .	2	25,130	10·01	1,630	0·6883	285·9	8·790	142·9	36·08
	Saving on	No. 1 test	2	—	—	6,407	—	—	22·90	—	2·620‡	—	—
5	Superheating	June, 1902	5	Duff	1	38,864	24·71	1,577	1·1001	281·8	13·79	56·4	22·31
	Loss on	No. 1 test	1	—	—	7,327	—	—	23·10	—	2·38§	—	—
‖6	Superheating	Feb., 1902	2	Small, with about 16 per cent. of rough nuts	1	19,656	11·86	782	1·125	211·6	9·29	105·8	28·21
	Saving on	No. 1 test	—	—	—	—	—	—	—	—	2·12¶	—	—

* A saving of 2·28 per cent. † A saving of 13·26 per cent. ‡ A saving of 22·96 per cent. § A loss of 20·8 per cent. ¶ A saving of 18·58 per cent. ‖ Pit only partly at work, i.e. Yard and New Blake Seams.

From the above it will be seen that with five boilers in use (No. 5 test), instead of a saving, there was a loss by using superheaters. With four boilers (No. 2 test) there was a saving of 2·28 per cent; with three boilers (No. 3 test), a saving of 13·26 per cent.; and with two boilers only (No. 4 test), there was a saving of 22·96 per cent.

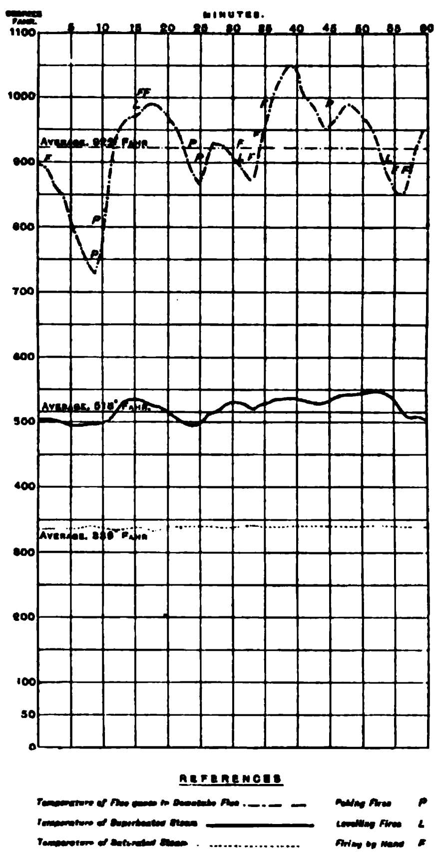


FIG. 280.

These facts, at first sight scarcely consistent with each other, are quite explicable when we bear in mind that with the same load, with the five boilers the fires would all be low, and the temperature of the flue gases low also, and the large extra surface of the five superheaters, as well as the extra range of piping, would naturally cause a loss rather than a gain.

As the number of boilers was reduced, the exposed surfaces would be less, and the temperature higher. The diagram subjoined (Fig. 280) shows that, with three boilers only (No. 3 test), the gases were seldom hotter than 1,000° and only averaged 922°, the average superheat being 518°. With only two boilers the fire and the flue would be still hotter, and the gain is proportionately greater. None of the engines used in connexion with these tests were compounded, otherwise the results would have been still better.

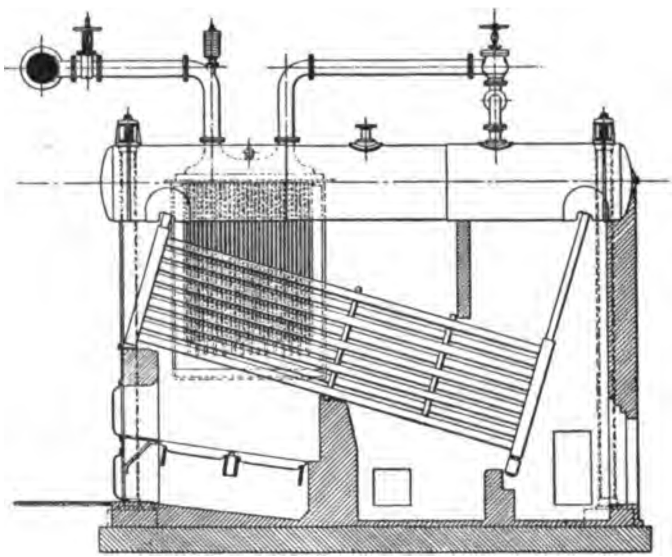


FIG. 280A.

Though we have shown the heaters only in connexion with Lancashire boilers, yet they can be applied to nearly all classes; and Fig. 280A shows a superheater in connexion with a tubulous boiler of the Babcock and Wilcox type.

There is some considerable analogy between the boiler and the engine, i.e. the steam generator and the steam user.

In order to get the most use out of steam of a high pressure we must have an engine with a high temperature cylinder, one or more intermediate cylinders and a low temperature part, i.e. the condenser. All these must be isolated from each other as far as possible; and, similarly, in order to get use out of a steam generator we must have a low temperature

part, i.e. the water heater separated as far as possible from the boiler, which is intermediate and a high temperature part, the superheater, again separated from the generator.

Again, just as a large compound or triple-expansion engine when worked at a very low power, is more wasteful than a smaller simple engine for the same work, so a large boiler with superheaters worked at a very low duty is less economical than a smaller simple boiler, with its great surface and dimensions proportioned to its work. In this, as is all other matters connected with engineering, in order to get the best results, the whole of the circumstances must be taken into consideration, and the plant in all its details must be proportioned to the work it has to do. In this way, and in this way only, can the user obtain the greatest satisfaction in using, and the greatest economy from using, his engine.

CHAPTER XVII

Hints to Users

WITH the great variety in the types of steam engines now manufactured, one desirous of using steam power may not unreasonably feel a difficulty in deciding which would be best for his purpose. This variety in the shape and construction of steam engines is as much due to the individuality of the designers as to any special construction to suit the different purposes to which they are applied. Year by year, however, these differences tend to get smaller, and the engines tend to conform to certain standard types which long experience has proved to be best. Many of the best examples of these standard types have been illustrated in chapter XIII dealing with turbines, and chapter XV dealing with types of engines.

The satisfactory and successful use of a steam engine depends so much upon being the one most suited for the purpose for which it is used, that too much care cannot be exercised in deciding upon the right one.

Normal Types. The first and perhaps the most valuable hint to the user is, not to be persuaded into experimenting with an abnormal and untried type of engine, however highly it may be recommended. None know better than experienced engineers how frequently engineering monstrosities are brought forward and advertised, and how even amongst good inventions when they are manufactured for the first time, some little thing may have been overlooked, neglected or miscalculated, resulting in trouble and disappointment to the user. The proper place in which to test the value of a new departure in engine construction, is in the manufacturer's own testing department, or in use on his own works for at least a year, and not in the hands of a customer. This practice may seem to cause delay in the introduction of improvements, but in the long run it is in the interest of both manufacturer and user.

It may be stated at once that the steam engine is not the only source of mechanical power available, though it now is, and for many years is likely to be, the most important and advantageous for the great majority of purposes. Of late years there has been a rush for driving everything by electricity, and vast sums of money have been wasted in putting down electric plant for the purpose which has served no useful end.

Electric Motors.

The writer has not infrequently come across cases where just because the current was available an electric motor has been put down to drive a tool when there was a revolving shaft in close proximity, from which power could have been taken quite as conveniently and much more economically. In some cases even in large factories the engines and boilers have been dismantled and in their place an electric supply from a public source has been installed at great cost for cable and motors, not only without any corresponding advantage but in some cases even at a great loss. Many steam engines now in use are certainly very wasteful in fuel, but in such cases it is a serious and important question to decide whether to replace them by more economical engines or to adopt electricity. Having decided that question in favour of electricity, the next question is whether to buy it from a public source or to make it in the factory. If required in large quantities, i.e. anything over 200 H.P., the private manufacture is more economical than purchase. So far as the cost in fuel is concerned, which is the principal item in a large plant, this is not more than $\frac{1}{8}$ of the cost of electricity, even when so low as $1\frac{1}{2}d.$ per unit. The charge for labour, superintendence, as well as the capital charge is, however, much larger relatively in the small plant than in the large one.

It must be remembered that in nearly every case electricity is first got from the power of a steam engine, and that there is some loss in transforming that power into electricity. There is also a similar and equal loss in re-transforming the electricity into power when it is used as such, so that, other things being equal, electrical power driving cannot be so economical as direct driving from a steam engine. Thus for such a plant as a roller flour-mill, when each part of the plant is always at work and at the same pressure often night and day, and when the mills are all close to each other in a small space,

electric driving would be a certain source of loss when compared with the highest class of steam engines. On the other hand, in the case of a shipyard where the tools, many of them heavy ones, are widely separated from each other, sometimes by a mile or more, and besides are only intermittently used, the cost of mechanical transmission of power either by shafting or rope, or of steam transmission and a multitude of small engines, is so great that a central high-class engine with generators and electric transmission may, and often does, make a saving of fully 75 per cent. in the cost of driving the plant. No inflexible rule can be laid down, each case must be considered on its merits.

Without question where power is required in small quantities and at occasional times, whenever an electric current is available, there is no other source of power so economical and convenient as an electric motor. It is always ready, starts at a moment's notice, needs practically no attention when running, and all cost in connexion with it ceases the moment it is stopped working, though while at work, the power is being paid for at an extravagant rate, even if supplied at 1½*d*. per unit, which is considered a very low price.

For the few readers who may not know exactly what an electric unit is, we may explain that an electric current has volume, and intensity, like water pressure. Thus a current of water through a pipe may be a large volume and small pressure, or a small volume and a high pressure, and in the one case make up by its volume what it lacked in speed, or in the other make up by its speed what it lacked in volume, so delivering the same quantity of water in the same time.

Electric Units.

When we speak of the pressure or *intensity* of an electric current, we use the term "volt," thus we speak of a current of 120 volts. When we speak of the *quantity*, or volume of a current, we use the word "ampère." These are arbitrary words from the names of eminent electricians, to signify the things "intensity" and "quantity." A combination of these two, i.e. volts and ampères, is called a watt, from James Watt of steam engine fame; thus 100 volts × 10 ampères equal 1,000 watts; conversely, 100 ampères × 10 volts equals 1,000 watts, and so in any other proportion; volts and amperes multiplied together equal watts, and a

supply of 1,000 watts for one hour is what is known as a Board of Trade Unit, or generally one unit.

Now 746 watts are equal to one mechanical horse-power, so one unit is equal to 1·34 H.P.; that being so, 1½*d*. per unit is very nearly 1*d*. per H.P. (exactly ·89*d*.), and this fact is useful for approximate calculations.

With a large engine of say 500 H.P. the cost of fuel is by far the greatest part of the expense of running, being for a good engine about 7 cwt. of coal per hour or 3½ tons a day of ten hours. This at 14*s*. a ton would be 49*s*. a day. The cost of attendance, oil, waste, etc., would be covered by another 11*s*., making a total of 60*s*. per ten hours. 500 electrical H.P. bought at ·89*d*. per H.P. would be 445*d*. or 37*s*. 1*d*. per hour = £18 11*s*. per ten hours, or a little over six times the cost of a steam engine.

Even if the water has to be bought as well, which is not often the case, then 7,000 lb. or 700 gallons of water per hour, or say 8,000 gallons a day at 9*d*. per 1,000 gallons = 72*d*. or 6*s*., which added to the 60*s*. makes a total of 66*s*. or £3 6*s*. as against the £18 11*s*., the latter being still over five times the cost of steam.

The case is, however, altogether different when we deal with small power. The cost of attendance with a steam engine is then greater than that of the fuel, and if this attendance can be dispensed with so great a saving is effected that the extravagant cost of the power itself can be borne and still a great economy effected. Thus 5 H.P. by electrical motor occasionally used for a total of three hours a day is only 1*s*. 3*d*. per day of ten hours, which is much cheaper than it can be got by any other means, and the power of one man (which is sometimes as much or more than is needed) can be got for ¼*d*. a hour.

Towns which have no electric current nearly always have a good gas supply, and much of what is said above applies equally to gas engines, i.e. with gas at not over 2*s*. 6*d*. per 1,000 feet, a horse-power can be got for 1*d*. For small power again gas engines may be advantageously used and are quite reliable and need little or no attention when running.

Gas Engines. When as much as 20 H.P. is occasionally required, say for an average of five hours a day, then the gas engine compares favourably with a simple steam engine.

The gas engine would cost about 1*s.* 6*d.* per hour when it was being used, and nothing at other times=7*s.* 6*d.* a day. If, however, it was used for the whole day at that power then its cost would be 15*s.* Now the cost of fuel for a simple steam engine for the same power would be the price of half a ton coals, say 8*s.*, and the driver's wages, 4*s.* 6*d.*, a total of 12*s.* 6*d.* a day instead of 15*s.* The cost of plant and its installation would be no more for the steam engine than for the gas engine and if a simple combined engine and boiler like that shown in Fig. 257 were used little if any more space would be required.

Of late years much attention has been devoted to the design and construction of gas or internal combustion engines, as they are called, and also to suitable gas-making plant to be used in connexion with them, and it is claimed that such combined plants can compete in economy with high-class steam engines. When, however, it comes to putting down and managing independent gas generators as well as engines there is no saving effected either in space or attendance as compared with steam, and in spite of the late improvements, internal combustion engines have not yet reached that simplicity and ease of management, together with that entire reliability which so strongly marks the use of steam. It is probably in the direction of internal combustion turbines that the best use of gas may ultimately be found.

For the present, unless it be a case where little or no suitable water can be found, or where the by-products from the gas-making plant are of special service or value, even granting all that is claimed for the gas plant, the balance of advantage still remains with the high-class steam engine.

Oil Engines. Oil engines are of similar type to gas engines. To all intents and purposes they are gas engines which manufacture their own gas within themselves at each cycle of revolutions. They are of great value when water is scarce and other fuel difficult to obtain. They were at first, and for some years remained, very capricious in working. Their construction has, however, been so much improved that now they can be depended upon with almost the same certainty as gas engines, and much of what has been written about the latter applies equally to the former. It is only a very careful investigation of all the circumstances which will

enable any one to decide intelligently what form of power it is best to use in each particular case.

Confining ourselves now to steam engines, we have to decide upon the type, and this very largely depends upon the

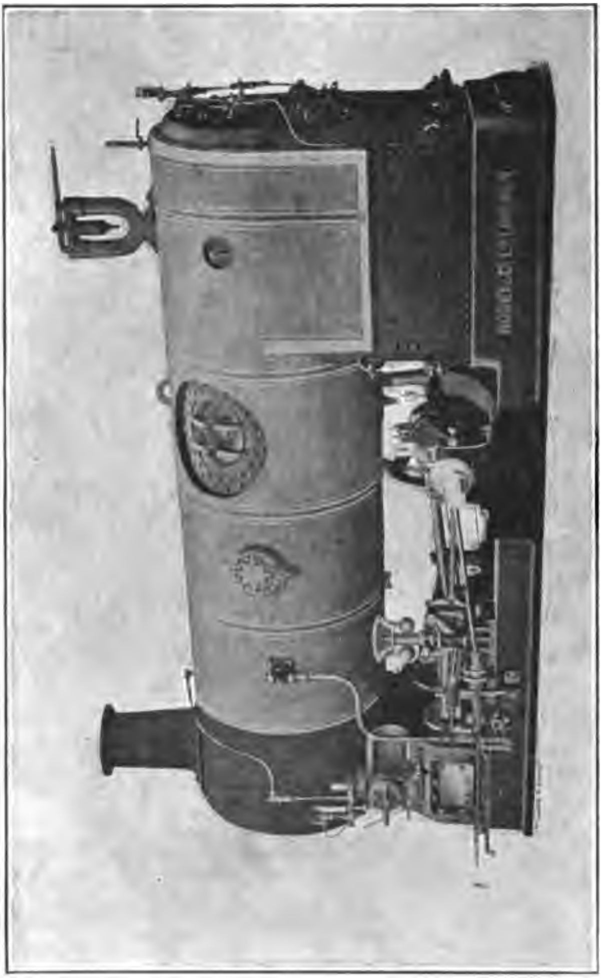


Fig. 257.

purpose for which it has to be used. A builder or contractor, needing power for short periods for many purposes, would use the portable engine (Fig. 256), one or a number of them. These are so simple in construction and management that

very often the man who works or helps to work one of the tools driven by it can also look after the engine. A portable engine can be delivered on a site and set to work within a couple of hours if there be anything of a water supply.

The type of steam engines used in agriculture for thrashing, hauling and ploughing are so well known and thoroughly proved that any description is needless.

Small Industries. For small industries needing 20 to 200 actual H.P. the combined engine and boiler shown in Fig. 257 is the most convenient and economical, especially when the business premises are held on lease for a short term of years. Such a type of engine does not need securing to the foundations and therefore remain the property of the tenant user.

There are many cases when it is not wise to seek after the highest economy in fuel; for instance, when this is very cheap. For a sawmill in or near a forest, sawing up tree-trunks into boards a simple high pressure engine and the simplest form of boiler is the best. There is here no question of saving fuel. The sawdust as well as the rough wood from the mill has to be got rid of in some way or payment may have to be made for its removal; it is evident, therefore, that the best way is to burn it in order to make steam, and there is always much more than enough for this purpose.

On the other hand, a saw mill in or near a large town in a thickly populated county is on quite a different plane. It does not make so much waste to begin with, then the sawdust or some of it may be sold for more than its value in fuel. Nearly all the wood sawn will be put to some useful purpose, for there are a thousand ways in which it can be utilized. All that is left then of shavings and small chippings need to be carefully used for fuel, and if a very economical engine be used this will be found to be nearly, if not quite, enough in many cases. These are the sort of facts which have to be borne in mind when each case is decided upon its merits.

Factors for Economy. The factors which tend to the highest economy, as we have already shown, are High pressure of Steam, Great Expansion, High Temperature of the feed water and the Superheating of the Steam. Whether any or all of these expedients shall be adopted is the matter for consideration according to the special circumstances.

There are, however, two things which should be carefully attended to whatever form or type of steam engine is employed, viz. the prevention of leakage, whether of heat or of steam. To prevent the first, all exposed surface, both of boiler and engine, must be carefully and

Ordinary Losses.

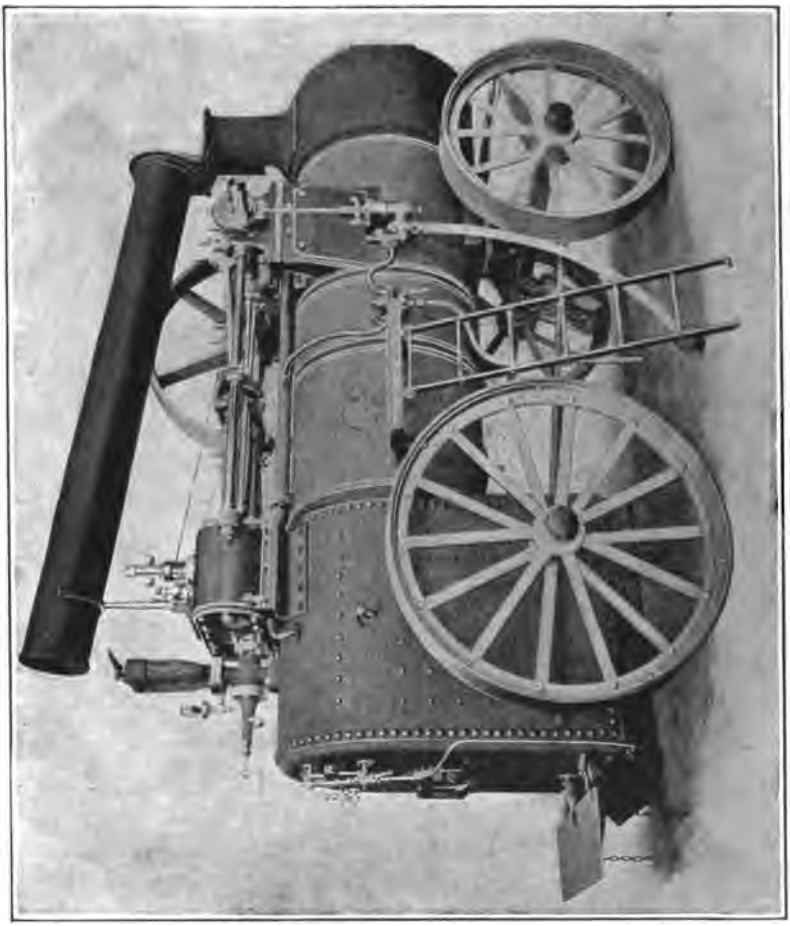


Fig. 256.

efficiently covered with some non-conducting material to such a thickness as to prevent any escape of heat. Leakages of steam are generally from bad joints in the steam pipes, and joints give way from the strains to which they are exposed. A length or range of steam piping ought to be at rest whether

hot or cold, and that it may be it must be free to expand and contract. The expansion of pipes by heat of the condensed steam is irresistible, and if it be fixed in a straight unyielding line its expansion and contraction will move the engine on its foundation or the boiler on its seat, and the pipe will soon begin to leak however well the joints are made. It is generally easy to arrange a range of piping with one or more bends as in Fig. 281. Pipes so arranged may be rigidly fixed at the ends, but unlike the straight pieces, are free to expand in the direction of the arrows when steam is turned into them. When from any good reason a straight pipe must be used what is known as an expansion joint is sometimes used; this is simply a stuffing-box and gland, one part of the pipe being turned to slide in and out of the gland and stuffing-box which is attached to the other part of the

Pipe Connexions.

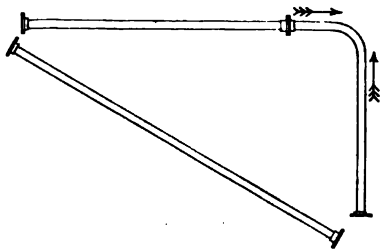


Fig. 281.

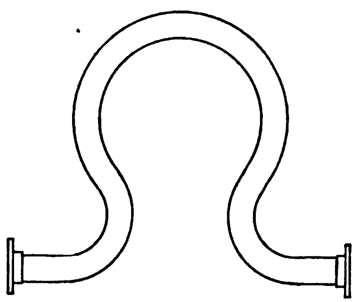


Fig. 282.

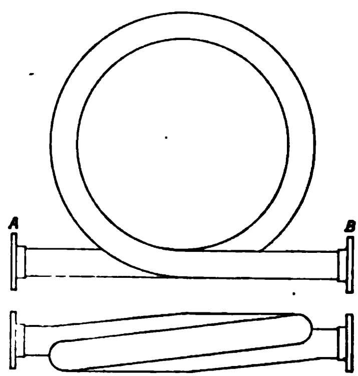


Fig. 283.

pipe. This is a very unsatisfactory method of allowing for expansion, for if the stuffing-box is screwed up tight enough to prevent leakage it will be too tight to prevent easy expansion and vice versa. Such devices are now seldom used, but instead some steam-tight device in the way of giving flexibility to the pipes such as a corrugated enlargement made of copper which

can expand and contract, or a bent pipe of copper as shown in Fig. 282. These are much better; they certainly admit of expansion without leakage, but they allow the pipe to increase in length when hot, which needs some other device to prevent the distance between the ends of the pipe altering their position. The author some thirty years ago devised a connexion shown in Fig. 283 which he has used in a very large number of installations with perfect success, and it is very simply made. This is an ordinary length of steel pipe bent round in a circle which should not be less than ten times the diameter of the pipe. This expansion ring has the great advantage that as it becomes hot and expands, the distance between the two flanges becomes shorter instead of longer, thus taking up the extension of the straight parts. This expansion ring, though largely used, is, so far as the author knows, now for the first time published.

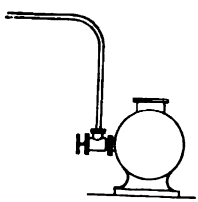


FIG. 284.

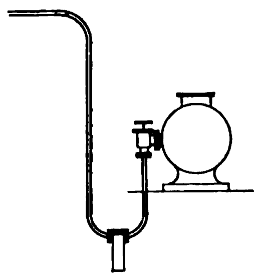


FIG. 285.

It is quite as essential to have the steam pipes covered with non-conducting material as the exposed parts of engine and boiler.[1]

When the steam is not superheated great care must be taken to see that the steam pipes are thoroughly drained; it is indeed a wise precaution to do so with superheated steam as the apparatus may be temporarily out of order and then the drainage will be needed. As far as is possible the pipes should be fixed on a slight incline so that they drain back automatically into the boiler, but this cannot continue for the whole distance. When the pipes bend down to the engine stop valve, they should not connect direct to it as in Fig. 284, because whatever water was in the pipe would be drained into

[1] For illustration of surface radiation losses *see* p. 55.

the cylinder. The best plan is to bring down the steam pipe below the floor line and up again to the engine stop valve as in Fig. 285, and at the bottom part of bend fix an automatic drain valve and pipe.

The lubrication of main bearings of a horizontal engine have been already dealt with on p. 280. The other important parts are the crank-pin and the cylinder.

Lubrication. It is highly important that every part of an engine should have provision for ample lubrication for long periods, i.e. a week or more; even if an engine be ordinarily stopped two or three times a day it may nevertheless happen that occasionally it may be found necessary to run on without stopping, but this cannot be done if it runs short of oil. The crank-pin is the part which is most difficult to deal with, but

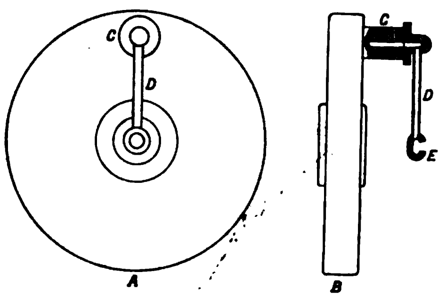


Fig. 286.

the difficulty is quite overcome by the centrifugal oiler shown in Fig. 286. Here *A* is the front elevation and *B* the side elevation of a driving disc and crank-pin, the latter being shown in section. In the crank-pin *C* a hole is drilled up the centre, from which hole one or more branch holes are drilled to the circumference of pin furthest away from centre of shaft. The tube *D* connects the cap of crank-pin with the central hollow ball *C*. The oil-cup being stationary can be replenished at any time, but can easily contain enough for a day's consumption. This oil-cup is so constructed that it delivers its oil drop by drop as required into the hollow ball *E*, the centrifugal force drives the oil to the inner circumference of the ball, and thence up the tube *D* into the crank-pin and out to its circumference.

The oil-cup should be one known as a visible feed, so that the

attendant may be able to see always that it is working properly.

For the cylinder there are now exceedingly good oils manufactured as well as a great variety of lubricators. Some have a positive feed and are worked by a ratchet and lever from some running part of the engine, or they are of the standard sight feed type delivering drop by drop. Whatever type of cylinder lubricator be adopted it must be one which gives a continuous feed, for oil put in by the cupful at long intervals, as was the custom formerly, is mostly wasted and passes away into the exhaust at once, while between times the cylinder may run dry. The best place in which to deliver the cylinder oil is between the stop valve and the valve chest; by so doing the oil gets pulverized and more thoroughly mixed with the steam, the valve as well as the cylinder is lubricated and the minimum quantity of oil is used.

Indicator. One of the most useful and important things for the steam engine user is the indicator. This has been partly dealt with in chapter V on valve gear, but its construction and use will now be more fully explained.

This, like many other important things, was an original invention of James Watt, and though it was only in a very primitive form at that time it served very well for the slow-speed engine then in use. Its construction and use was kept secret by Watt for many years, and even up to comparatively modern times the indicator was looked upon as a sort of mystery. As a matter of fact it is extremely simple. Many different varieties are made which give accurate and satisfactory results, one of the best of which for any but very high speeds is the Richards indicator shown in the illustration (Fig. 287).

A is a cylindrical casing containing a steam cylinder with the piston *B*, a spiral spring *D* is screwed at one end to the boss of the piston and at the other end to the cover *E*. The piston-rod works through a hole in the centre of the cover *E*, by which it is guided in a straight line. A short connecting link connects the piston-rod to the lever *G*, which is pivoted on the fixed point *H*. A corresponding lever of equal length is pivoted at *M* and the outer ends of these two levers are connected together by the motion bar *K*, the whole forming a parallel motion like that described in chapter III at Fig. 18.

The central point *K* in the bar has a metallic pencil fixed in it and it travels in a straight line. The inner nut *U* connects the cylinder of the indicator with the cock *T*, which in its turn is in direct communication with the inside of the cylinder of the engine being indicated, and the piston *B* is exposed to the same pressure as the piston of the engine. This pressure is resisted by the spring *D*, and these springs have to be very carefully made in order to allow for the very different pressures of steam used in the engine cylinder or cylinders. These springs are made in different standard sizes and strengths suitable for any pressure of steam from 10 lb. to 200 lb. above the atmosphere and 15 lb. below it.

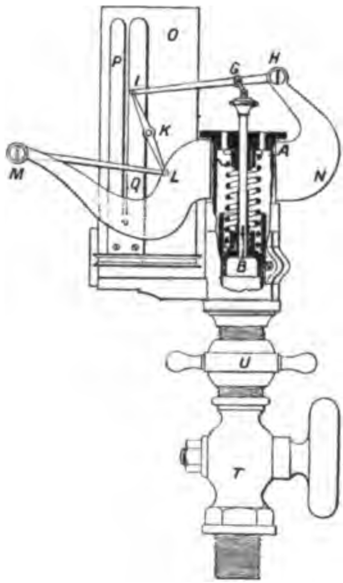


Fig. 287.

It is advisable always to use a spring a few pounds stronger than the pressure for which it is going to be used, so that there may be no risk of driving the piston up to the top of its cylinder, which would make it give an inaccurate result. The extreme travel of the pencil of a Richards indicator is 3 inches in height, but it is advisable to keep the height of the diagram well within the limit. On the other hand, the spring should not be too strong so as to unduly limit the height. If, for instance, we used a spring of 100 lb. to the inch for steam of 25 lb. we should get a diagram only ¼ of an inch high, which would be practically useless.

Having now clearly seen how the pressure is indicated by the pencil *K* we next notice how it is recorded. A small barrel *O*, 2 inches diameter, is arranged to work freely on a central spindle. In the lower part of the barrel there is a coiled spring like a clock spring, this is made very elastic, but sufficiently strong to pull tightly at a string which is coiled round the groove shown at the bottom of barrel *O*. This string is shown in the outside view of the indicator (Fig. 287A),

where the cord is shown in position between two little guide pulleys. These keep it in place on the barrel while allowing it to be directed up or down. The plate containing the guide pulleys also swivels round, and can be fixed in any position, and thus the cord may be pulled in the direction that is most convenient. This cord needs to be fastened to some part of the engine moving simultaneously with the engine piston and preferably in the same direction.

FIG. 287A.

There are two spring clips, *P* and *Q*, which are for the purpose of holding a sheet of paper in close contact with the barrel. The paper must be long enough to wrap round the barrel and to leave ends projecting about ⅜ of an inch through the slot between the clips, so that it can be taken hold of by the fingers and pulled tight on to the barrel. The paper should be a little less in width than the length of barrel. A piece is shown in position in Fig. 287A.

The barrel cannot make a complete revolution, as stops for the purpose prevent it doing so, but it can go nearly round in one direction, being pulled by the cord, and then back again, being pulled by the spring. That being so, if the pencil at *K* is held lightly against the paper on the barrel it would make a straight horizontal line when the piston *B* has no steam pressure under it, and only the atmosphere pressure above it. Such a line is called the atmospheric line and should always be made on a diagram. Similarly, if the barrel be at rest and the indicator cock *T* be open so that it is put into communication with the engine cylinder, then the point *K* would travel up and down in a straight line at right angles to the atmospheric line.

When, however, the barrel and the piston of the indicator are both moving, then a continuous line will be traced on the paper, the height of which will vary with the pressure of the steam in the engine cylinder, and the length of which will correspond with the travel of the barrel. Such a line is what

is called a steam engine diagram, an example of which is shown in Fig. 288. The circumference of the barrel is from A to B, the parts beyond being the overlap of the paper come between the spring clips P and Q. The full travel of the barrel is from C to D, and the height of vertical movement of the piston is from C to E and D to F. The diagram occupies the space in the centre, its outline being well removed from the extreme travel in either direction.

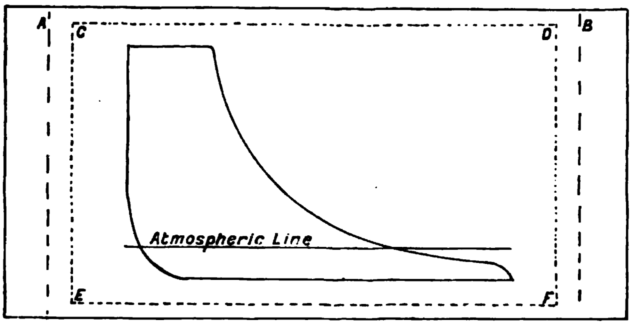


FIG. 288.

It is always best to leave about this amount of margin, because, as stated before, any approach to the extreme limits is apt to lead to inaccuracies.

Reducing Gear. One of the simplest methods of making the string actuate the barrel is shown in Fig. 289. Here a lever H swings on a fixed pivot K; the latter, though not exposed to any great strain, yet must be quite rigid. This lever is used for the purpose of reducing the travel of the barrel under the pencil point K to a little less than that of the barrel's circumference, say from the stroke of the engine 48 inches, to 4 inches, the length which we require our diagram to be; this is $\frac{1}{12}$. Thus the radial segment L, which is fixed on the boss of the lever H, must have a $\frac{1}{12}$ of the radius of the line or 4 inches. This segment can be adjusted on the lever boss by a set screw, so as to lead the cord off in any direction; the cord must always be a tangent to its circumference. Such an arrangement is very simple, and is what is frequently adopted.

If the lever H is made of a length equal to, or greater than the stroke of the engine, and the small connecting-rod is not less than 9 inches for a 4-foot stroke, the motion will be nearly,

but not quite accurate. It will be seen on referring to Fig. 289, that the path of the crosshead is from *A* to *B*. If the radius rod *H* could be of infinite length, then the radius shown in the curved dotted lines would be a straight line, and its junction with the connecting-rod *E* would lie in a straight line from *C* to *D*; but, owing to the curve it makes, and, still more, to the curve made by the connecting-rod *E*, the real path is from *I* to *J*. The points *I* and *J* are both to the right of the true points, and consequently towards *I* the rod travels more slowly, and towards *J* more quickly than it should do. This tends to make the admission line at *C* end, a little longer, and

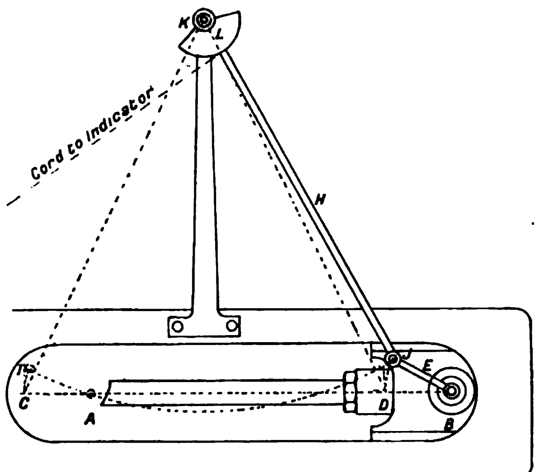


Fig. 289.

at *D* end, a little shorter than it really is. A much better arrangement and one equally simple is that shown in Fig. 290. In this the radius rod is of the same length, i.e. equal to the stroke, but instead of having a connecting-rod, there is a parallel slot in the end of the rod; the pin on the end of cross-head engaging in it as shown. The path described by the rod, though still slightly inaccurate owing to its varying length, is yet perfectly equal at both ends; the indicator cards will therefore be equally true at each end. For everything excepting the most refined investigations, this method is as good as can be. The cord to the indicator barrel is taken off the segment as shown, and may be led in any direction.

Having now obtained a correct method of taking diagrams, the question arises for what purpose does a steam user need it ? For two things principally. First, for seeing that the valve gear is properly made and properly set, i.e. that the admission, cut-off, exhaust and compression are what they should be ; second, for ascertaining the power the engine is giving off, and for checking this with the full limit. If the steam user wishes for the greatest economy in working, he must do this periodically and carefully. The amount of extravagance and waste which still goes on in connexion with many engines is enormous.

Object of Indicating.

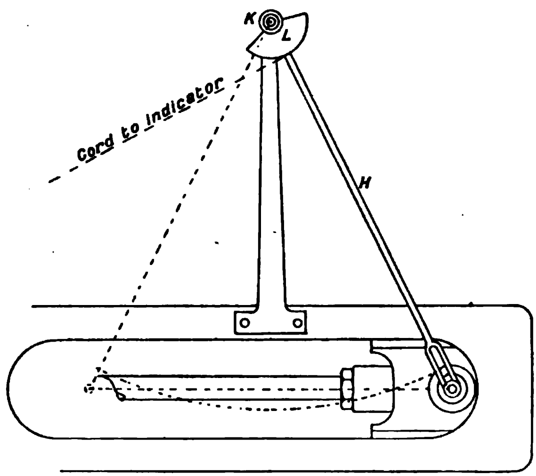


Fig. 290.

It is not too much to say that, taking steam engines as a whole, at least twice as much coal is burnt as is necessary to produce the power got from them.

There is generally a large margin of power in steam engines, and they will go on working smoothly with defects in their installation and steam distribution, which result in most extravagant waste.

Many of these defects can be easily and cheaply remedied. It is impossible within the limits of this chapter to point out all, or even many of the faults, but a couple of sample and extreme cases may be dealt with. One is a case of late and sluggish admission and excessive inside lap ; a diagram (Fig. 143) for the

engine has already been dealt with in chapter IX. Here the steam was admitted late, and was exhausted so late owing to inside lap, that it only began to exhaust at C; the cylinder was at that time filled with steam which could not get freely away. It made, therefore, a large back pressure, reducing

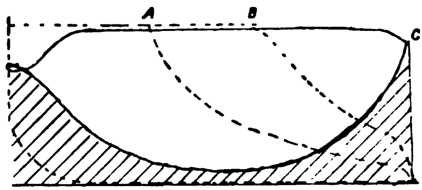


FIG. 143.

the power of the engine; the compression was excessive. The shaded part shows the back pressure. By simply altering the lead of the valve and removing the inside lap, the consumption of steam was reduced by fully $\frac{1}{2}$. With corresponding power the engine then cut off at A, and had a margin from A to B for still larger power.

Solution of Problem.

This was a bad example, or rather a good example, of bad condition, but the following is still worse:—

An engine of 100 H.P. was purchased from a very good maker and fixed on the purchaser's premises by a local engineer. He, to suit its placement, thought it advisable to reverse the direction of its running; this was done, and the engine was started to work. It was a double-cylinder engine fitted with slide admission valve and cut-off valve on the back, worked by a separate excentric. For a time all seemed to go on fairly well, but, as the load was gradually increased, a difficulty was experienced in keeping the boiler supplied with water. The local engineer said the feed pump was not sufficiently large. Complaint was made to the maker who assured the users that the pump was large enough to supply 50 per cent. more water than the engine should use in the form of steam. In the difficulty the author was called in to investigate and report to the owner. The pump by all calculation was ample for the engine, and was in good working order, but there was the fact that the load had to be thrown off the engine two or three times a day to enable the boiler to be filled up. Of course a supplementary pump for the boiler

would have been a simple remedy and one with which the user would have been satisfied, but that did not get to the root of the matter, nor was it satisfactory to the maker of the engine. The first thing to do was to indicate the engine, and the card obtained is shown in Fig. 291. This looked at first sight like an admission of steam at *A* and a cut-off at *B*, with an exhaust almost immediately after. One thing was certain, that the valve gear was wrong, for there was a total absence of any sign of steam expansion. Another card was taken of the same kind, but then the fact was revealed that the admission end of the card was at *C*, and the exhaust end at *A*, the nearly vertical line terminating at *B* was the admission line, the exhaust taking place at *A*; thus the steam

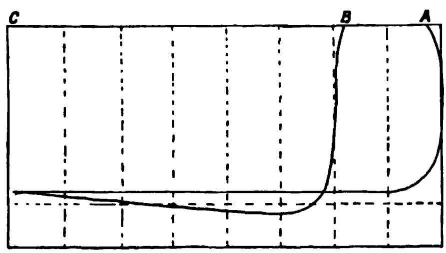


FIG. 291.

was not admitted until $\frac{3}{4}$ of the piston stroke had taken place. Thus the whole cylinder was filled with steam at nearly boiler pressure, which was soon after thrown away in the exhaust.

It is difficult to conceive of a much more wasteful expenditure of steam, and the mystery of the shortness of water in the boiler was at once explained.

An examination of the valves showed at once how this had taken place. The local engineer had rightly changed the position of the main valve excentrics by turning their full side from one side of the crank to the other, and had then turned the cut-off excentric from its position nearly opposite to the crank-pin, to a position nearly coincident with it. The result of this action was that no steam could be admitted into the cylinder at the beginning of the stroke, as the admission port in the main valve remained covered until $\frac{3}{4}$ of the stroke was done. It was a simple matter to put right. The cut-off excentric was put into its right position, and the engine

set to work again; when, with the same load, the diagram (Fig. 292) was obtained, showing a cut-off at $\frac{1}{8}$ with a considerably greater load upon the engine; thus the greater load was done with only $\frac{1}{8}$ of the steam formerly used, and the total saving in the fuel burnt was over 80 per cent.

Margin of Power and Economy.

Further, the diagram showed that there was a great margin of power in the engine; it being, with a diagram like Fig. 292, working at less than its most economical power.

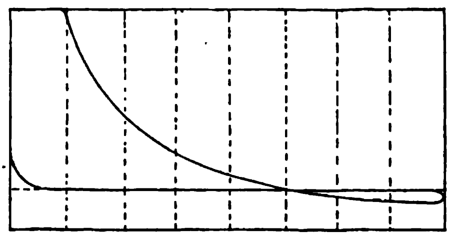


Fig. 292.

It is not to be expected that many investigations would result in such a saving, but, nevertheless, there are, without doubt, hosts of steam engines working in such conditions that a very large percentage of the steam is wasted, and there is no better way of finding out these defects than by the engine indicator.

For very high pressures and high speeds an indicator with still lighter recording gear is used. Amongst the best of these is that known as the New Crosby, and shown in Fig. 293. There are two or three important departures from the standard type in this indicator. The cylinder is made of 1 inch area instead of $\frac{1}{2}$ inch, and the piston instead of being about $\frac{1}{2}$ inch deep, is very narrow and touches the cylinder only at a line, its form being that of a slice cut from the central part of a sphere; there is thus comparatively no friction. The spring is removed altogether from the piston and is carried well above the recording gear; it is thus free from the heat of the steam, and in a much more convenient position for changing the springs. The lever which carries the recording point is extremely light; hence the indicator is very sensitive to rapid changes, and is especially suitable for high speeds.

Simultaneous Diagrams. In testing an engine having several cylinders it is sometimes requisite to take diagrams simultaneously from each end of each cylinder, and for this purpose Sargent's Electrical Attachment will be found of great value. This consists of a small electro-magnet so arranged on the indicator that the detent which holds the paper barrel stationary, is released the instant an electric current is passed

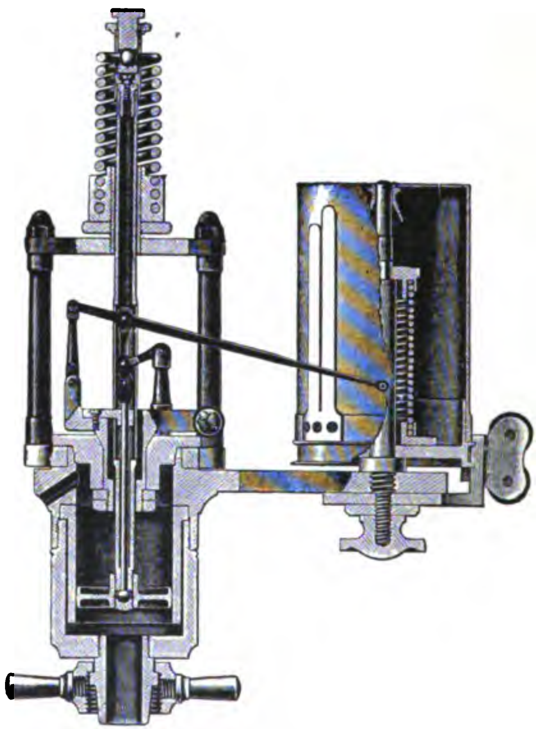


FIG. 293.

through it; by touching a switch and closing the circuit of the current, all the indicators are set to work at once.

Reducing Gears. There are many other reducing gears besides the ones illustrated in the preceding pages; and so long as the motion of the piston is correctly reduced, and no slack or back lash in the cord is introduced, the method may be varied indefinitely, and must often be modified to suit special circumstances. How the power of the engine is

calculated from the diagram has been fully dealt with, (pp. 47 to 52, chap. IV), and need not be further gone into here. The foregoing remarks are intended to show to users the great use the steam engine indicator may be to him if he desires to get the greatest power out of his engine with the smallest expenditure for fuel.[1]

So far we have dealt in this chapter principally with existing engines. In putting down a new engine there are many matters which have to be taken into consideration before its type is decided upon. Amongst these are the water supply, the cost of fuel, the space available, and the kind of machinery to be driven. Engines most suitable for driving the delicate quick-running machinery of a cotton or silk mill, would be quite out of place in a brick and tile works where heavy machinery has to be driven at a slow speed. The nearer the engine approximates in speed to that of the machinery driven by it, the less will be the loss of power in transmission.

Fuel and Labour.

Again, the relation the engine power bears to the amount of labour employed is important. Thus in a well-designed roller flour-mill, sometimes from the unloading of the grain from the ship to the deliverer of the flour into sacks—each one weighed accurately—the whole process is automatic, requiring little or no labour. There the very highest class of engine with every refinement for securing the utmost economy should be used, as the cost of fuel is the principal item in the cost of production.

It is quite otherwise in a clothing or boot factory crammed full with many hundreds of small and delicate tools, each with a skilled worker in constant attendance. There, certainty and regularity are the prime considerations, and some sacrifice in economy of fuel may be reasonably granted, if by so doing the risk of stoppage for a single moment can in any degree be removed. The cost of fuel in this case being so small in proportion to the labour bill.

It is manifestly impossible to lay down any definite rules

[1] For the student or user who wishes to go very fully into the matter of indicating and the peculiarities of indicator cards, *The Indicator Handbook*, by the Editor of the *Mechanical World*; and *Practical Instructions for Using the Steam Engine Indicator*, published by the Crosby Steam Gauge Co., Boston, U.S.A., will be found of great utility.

the following of which will always secure success. There are special circumstances connected with nearly every case, and these must be specially considered. Nevertheless, the author trusts that a study of the contents of this book will fulfil the purpose indicated in the preface and will materially help to teach all who design, construct or use steam engines, What to avoid, What to do and How best to do it.

Appendix

THERE are a number of recognized and well-established laws relating to Heat or Thermodynamics which it is well to know. Amongst the most important are the following:—

FIRST LAW OF THERMODYNAMICS.—"Heat and mechanical energy are mutually convertible, and heat requires for its production, and produces by its disappearance, a definite number of units of work for each thermal unit."

In other words, "when work is done by means of heat, a definite quantity of heat is lost for every unit of work done." And conversely,

"When heat is produced by the expenditure of work or energy, an equal and definite quantity of heat comes into existence for every unit of work spent."

THE SECOND LAW OF THERMODYNAMICS.—"Heat cannot pass from a cool body to another body hotter than itself by a purely self-acting process."—(Clausius.)

UNIT OF WORK.—According to the experiments of Joule it was found that it takes 772 (since corrected to 778) foot pounds of work to raise one pound of water one degree (F.) in temperature, starting at 39° F. So 778 is called a "Thermal Unit."

LAWS OF PERMANENT GASES.

Boyle's Law.—(1) "The volume of a given mass of gas varies directly as the pressure, provided the temperature is kept constant."

Thus pressure and volume multiplied together always give the same result.

Charles' Law.—(2) "Under *constant pressure* equal volumes of different gases increase equally for the same increment of temperature. Also, if a gas be heated under constant pressure, equal increments of its volume correspond very nearly to equal intervals of temperature as determined by the scale of a thermometer."

Regnault's Law.—(3) "The specific heat and constant pressure is constant for any gas."

In accordance with above law experiments show that the amount by which a gas expands when its temperature is raised by 1° F. (pressure being kept constant) is about $\frac{1}{493}$ of its volume.

By "specific heat at constant pressure" is meant the heat taken up by 1 lb. of a substance when its temperature is raised 1° F., the pressure remaining the same, while the volume changes. Thus if 493 cubic inches of air or any permanent gas be taken at the temperature of 32° F. (freezing-point), and it be heated to 33° the volume will have been increased by $\frac{1}{493}$, and it will now be 494 cubic inches; at 35° it would be 496, and so it would go on increasing in volume one cubic inch for each degree, until with a temperature of 539°, i.e. 32° + 507°, its volume would be 493 + 507 = 1,000 volumes, the pressure remaining constant.

Conversely, the volume would fall with corresponding falls in temperature; thus, there being 493 volumes at 32° (freezing-point), then at

30° there will be only 491 volumes, at zero F. only 461, and therefore it is reasoned the absolute zero must be – 461; at which temperature, or rather absence of temperature, the gas would have no volume, and therefore no existence, which is absurd. (This third law of Regnault forms a good working hypothesis and is approximately correct within wide limits, but in the author's opinion there is much more to be learned about very high and very low temperatures, and when that knowledge has been obtained we shall have a more precise law.)

In practice, long before – 461 could be reached, our gas would first have become liquid and afterwards solid. The pressure being constant, it could not decrease in volume any further, even at – 461°. So long, however, as the gas remains a gas the relative temperatures and volumes with constant pressure are practically correct.

Joule's Law.—(4) When a gas expands without doing any external work, and without taking in, or giving out any heat (therefore without changing its stock of internal energy) its temperature does not change. Thus if we have a volume of gas contained in a cylinder one foot in area and two feet stroke, with the piston at half-stroke, it would have a volume of one cubic foot at say a temperature of 300°. If now, by any mechanical external means we raise the piston to the end of its stroke, so that the volume below it was increased to two cubic feet, the temperature would remain at 300°; but if we load the piston and the pressure of gas within the cylinder raised the load till its volume became two cubic feet, then work will have been done by the gas, and in proportion to the heaviness of the weight raised, i.e. the amount of work done, the temperature of the gas will fall. The diagram, Plate V, gives a graphic representation of the increase of temperature with increase of pressure. A scale of pressures from 0 to 1,000 lb. is shown on the thick black line on the top of the diagram. A scale of temperatures from 0 to 600 degrees Fahrenheit is shown on the thick black line to the left and right of diagram.

The black dots show temperatures, as they are given in standard tables. These are correct within the limits of practical observation, i.e. from 5 to 200 lb. pressure; above 200 lb. they are slightly irregular, but the curved line drawn gives a more regular and accurate indication and, as is seen, corresponds exactly with some. It is from this curve that the following table of pressure and temperature has been taken. The diminishing rate at which temperature rises is clearly shown.

TABLE I

PROPERTIES OF SATURATED STEAM

Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.	Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.	Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.
·125	24·5	·0001	—	2	126·3	·0058	10,721	10	193·9	·0264	2,358
·250	45·0	·0006	—	3	141·6	·0085	7,322	11	198·4	·0289	2,157
·375	61·2	·0011	—	4	153·0	·0112	5,583	12	202·5	·0314	1,986
·500	73·2	·0016	—	5	162·2	·0138	4,527	13	206·4	·0338	1,842
·625	83·5	·0020	—	6	170·2	·0163	3,813	14	210·0	·0362	1,720
·750	92·1	·0024	—	7	177·2	·0189	3,298	15	213·4	·0382	1,610
·875	97·9	·0027	—	8	183·4	·0214	2,909	16	216·6	·0406	1,515
1	102·1	·0030	20,582	9	188·9	·0239	2,604	17	219·6	·0430	1,431

TABLE I.—(*continued*).

Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.	Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.	Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.
18	222·5	·0454	1,357	**71**	303·7	·1670	373	**124**	343·2	·2822	220·5
19	225·3	·0478	1,290	**72**	304·6	·1692	368	**125**	343·8	·2845	219·0
20	228·0	·0507	1,229	**73**	305·5	·1714	363	**126**	344·4	·2867	217·5
21	230·6	·0525	1,174	**74**	306·4	·1736	359	**127**	345·0	·2899	216·0
22	233·1	·0549	1,123	**75**	307·3	·1759	353	**128**	345·6	·2911	214·5
23	235·5	·0573	1,075	**76**	308·2	·1782	349	**129**	346·2	·2933	213·0
24	237·8	·0596	1,036	**77**	309·1	·1804	345	**130**	346·8	·2955	211·5
25	240·1	·0620	996	**78**	310·0	·1826	341	**131**	347·4	·2977	210·0
26	242·3	·0643	958	**79**	310·9	·1848	337	**132**	348·0	·2999	208·5
27	244·4	·0666	926	**80**	311·8	·1869	333	**133**	348·6	·3020	207·0
28	246·4	·0689	895	**81**	312·7	·1891	329	**134**	349·2	·3040	205·5
29	248·4	·0713	866	**82**	313·6	·1913	325	**135**	349·8	·3060	204·0
30	250·4	·0740	835	**83**	314·5	·1935	321	**136**	350·4	·3080	202·5
31	252·4	·0759	813	**84**	315·4	·1957	318	**137**	351·0	·3101	201·0
32	254·3	·0782	789	**85**	316·2	·1980	314	**138**	351·6	·3121	199·5
33	256·2	·0805	767	**86**	317·0	·2002	311	**139**	352·2	·3142	198·0
34	258·0	·0828	746	**87**	317·8	·2024	308	**140**	352·8	·3162	196·5
35	259·8	·0855	726	**88**	318·6	·2044	305·	**141**	353·4	·3184	195·0
36	261·5	·0881	704	**89**	319·4	·2067	301	**142**	354·0	·3206	193·5
37	263·2	·0905	688	**90**	320·2	·2089	298	**143**	354·6	·3228	192·0
38	264·8	·0929	671	**91**	321·0	·2111	295	**144**	355·2	·3250	191·0
39	266·4	·0952	655	**92**	321·8	·2133	292	**145**	355·7	·3273	190·0
40	267·9	·0974	640	**93**	322·6	·2155	289	**146**	356·2	·3294	189·0
41	269·4	·0996	625	**94**	323·3	·2176	286	**147**	356·7	·3315	188·0
42	270·8	·1020	611	**95**	324·0	·2198	283	**148**	357·2	·3336	187·0
43	272·2	·1042	595	**96**	324·7	·2219	280	**149**	357·7	·3357	186·0
44	273·6	·1065	585	**97**	325·4	·2241	277·5	**150**	358·2	·3377	185·0
45	274·9	·1089	572	**98**	326·1	·2263	275·0	**151**	358·7	·3398	184·0
46	276·2	·1111	561	**99**	326·8	·2285	272·5	**152**	359·2	·3419	183·0
47	277·5	·1133	550	**100**	327·5	·2307	270·0	**153**	359·7	·3440	182·0
48	278·9	·1156	539	**101**	328·2	·2329	267·5	**154**	360·2	·3462	181·0
49	280·2	·1179	529	**102**	328·9	·2351	265·0	**155**	360·7	·3484	180·0
50	281·4	·1202	518	**103**	329·6	·2373	262·5	**156**	361·2	·3505	179·0
51	282·6	·1224	509	**104**	330·3	·2393	260·0	**157**	361·7	·3526	178·0
52	283·8	·1246	500	**105**	331·0	·2414	257·5	**158**	362·2	·3547	177·0
53	285·0	·1269	491	**106**	331·7	·2435	255·0	**159**	362·7	·3568	176·0
54	286·2	·1291	482	**107**	332·4	·2456	253·0	**160**	363·2	·3590	175·0
55	287·4	·1314	474	**108**	331·1	·2477	251·0	**161**	363·7	·3611	174·0
56	288·5	·1336	466	**109**	333·8	·2499	249·0	**162**	364·2	·3632	173·0
57	289·6	·1364	458	**110**	334·5	·2521	247·0	**163**	364·7	·3653	172·0
58	290·7	·1380	451	**111**	335·2	·2543	245·0	**164**	365·2	·3674	171·0
59	291·8	·1403	444	**112**	335·9	·2564	243·0	**165**	365·7	·3695	170·0
60	292·9	·1425	437	**113**	336·6	·2586	241·0	**166**	366·2	·3716	169·0
61	294·0	·1447	430	**114**	337·2	·2607	239·0	**167**	366·7	·3737	168·0
62	295·0	·1469	424	**115**	337·8	·2628	237·0	**168**	367·2	·3758	167·0
63	296·0	·1493	417	**116**	338·4	·2649	235·0	**169**	367·7	·3779	166·0
64	297·0	·1516	411	**117**	339·0	·2652	233·0	**170**	368·2	·3800	165·0
65	298·0	·1538	405	**118**	339·6	·2674	231·0	**171**	368·7	·3819	164·0
66	299·0	·1560	399	**119**	340·2	·2696	229·0	**172**	369·2	·3839	163·0
67	300·0	·1583	393	**120**	340·8	·2738	227·0	**173**	369·7	·3859	162·0
68	301·0	·1605	388	**121**	341·4	·2759	225·0	**174**	370·2	·3879	161·0
69	301·9	·1627	383	**122**	342·0	·2780	223·5	**175**	370·7	·3899	160·0
70	302·8	·1648	378	**123**	342·6	·2801	222·0	**176**	371·2	·3921	159·0

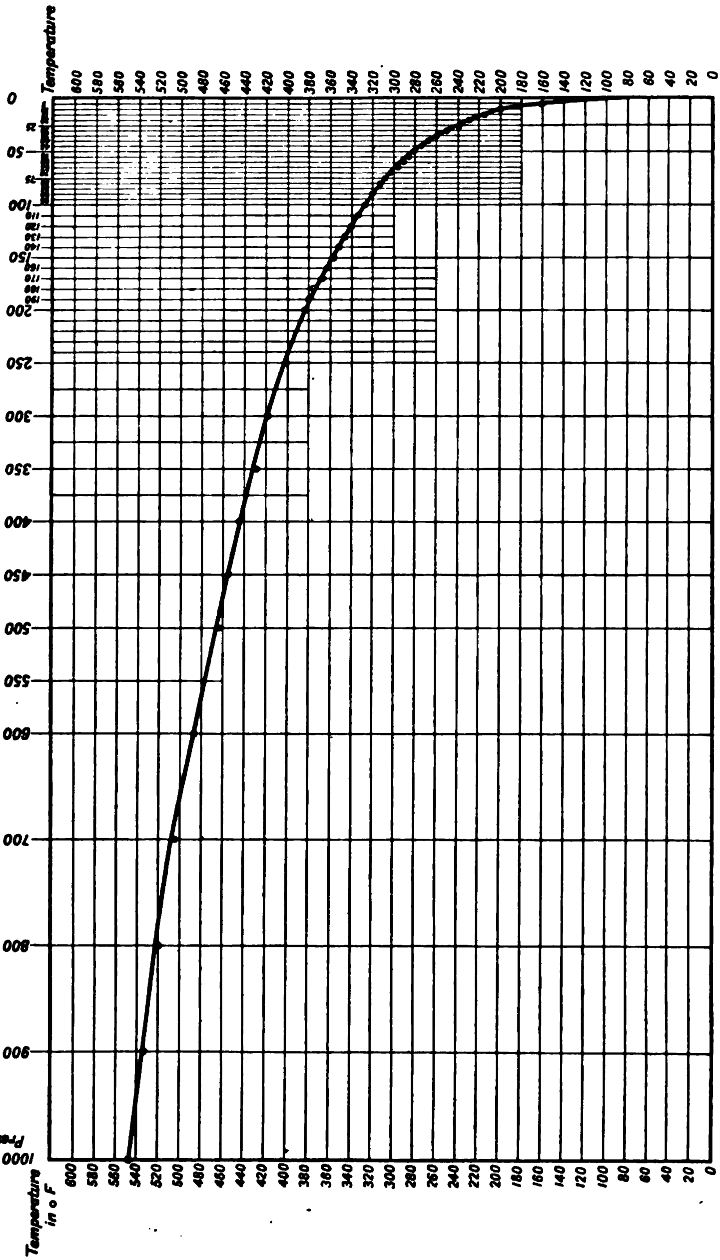
TABLE I.—(*continued*)

Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.	Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.	Pressure in lb.	Temp. in degs. F.	Weight of 1 cubic ft. in lb.	Specific Volume.
177	371·7	·3943	158·0	**215**	387·6	·4744	132·0	**390**	441·8	·8302	75·4
178	372·2	·3965	157·0	**220**	389·6	·4848	129·0	**400**	444·3	·8502	73·6
179	372·7	·3987	156·0	**225**	391·6	·4952	126·0	**425**	450·0	·9001	69·3
180	373·2	·4009	155·0	**230**	393·6	·5056	123·0	**450**	455·6	·9499	68·0
181	373·7	·4031	154·0	**235**	395·6	·5160	121·0	**475**	461·1	·9951	61·5
182	374·2	·4052	153·2	**240**	397·6	·5260	119·0	**500**	466·5	1·0502	57·9
183	374·7	·4073	152·4	**245**	399·6	·5364	117·0	**525**	471·8	1·0997	55·0
184	375·2	·4094	151·6	**250**	401·6	·5464	115·0	**550**	476·0	1·1492	52·1
185	375·6	·4115	150·8	**255**	403·2	·5566	113·0	**575**	481·1	1·1988	50·3
186	376·0	·4136	150·0	**260**	404·6	·5668	111·0	**600**	486·1	1·2482	48·6
187	376·4	·4157	149·2	**265**	406·1	·5770	109·0	**625**	491·0	1·2972	46·8
188	376·8	·4178	148·4	**270**	407·6	·5872	107·0	**650**	495·8	1·3462	45·1
189	377·2	·4199	147·6	**275**	409·1	·5974	105·0	**675**	500·5	1·3952	44·6
190	377·6	·4220	146·9	**280**	410·6	·6076	103·0	**700**	505·1	1·4442	43·2
191	378·0	·4241	146·2	**285**	412·1	·6178	101·0	**725**	509·6	1·4927	41·9
192	378·4	·4262	145·5	**290**	413·6	·6280	99·0	**750**	513·0	1·5412	40·7
193	378·8	·4283	144·9	**295**	415·1	·6382	97·5	**775**	517·2	1·5897	39·6
194	379·2	·4304	144·3	**300**	416·6	·6484	96·0	**800**	521·0	1·6382	38·1
195	379·6	·4325	143·7	**310**	419·6	·6688	93·0	**825**	524·9	1·6862	37·2
196	380·0	·4346	143·1	**320**	422·6	·6892	90·0	**850**	528·7	1·7342	36·4
197	380·4	·4367	142·6	**330**	425·4	·7094	87·5	**875**	532·3	1·7822	35·7
198	380·8	·4388	142·1	**340**	428·2	·7296	85·0	**900**	535·7	1·8302	35·0
199	381·2	·4409	141·6	**350**	431·0	·7498	83·0	**925**	538·9	1·8777	34·3
200	381·6	·4430	141·0	**360**	433·7	·7700	81·1	**950**	541·9	1·9252	33·7
205	383·6	·4535	138·0	**370**	436·3	·7902	79·2	**975**	544·6	1·9727	33·7
210	385·6	·4640	135·0	**380**	439·2	·8102	77·3	**1000**	547·0	2·0202	32·2

In addition to the temperature and specific volumes, Table I contains the weight of a cubic foot of steam at the various pressures. The weights of the greater part of them are calculated from Regnault's experiments, but the extremely high and the extremely low pressures are derived from a curve, based on the experimental data, and while therefore, only hypothetical, it is reasonable to suppose them approximately correct.

Various methods of finding the average pressures in a cylinder with different pressures of steam and various points of cut-off have been given in chapter IV., p. 47. All these methods require some time to be spent in calculation. In order to save this time the following very complete Table II has been given for reference.

[Plate V.



Curve showing Fall in Temperature at Different Pressures.

TABLE II

Expansion of Steam

5 lb. absolute. Specific vol., 4,527. Temp., 162·2° F.

5 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
$\frac{3}{4}$	4·82	3·75	150·6
$\frac{5}{8}$	4·56	3·12	143·0
$\frac{1}{2}$	4·23	2·50	134·0
$\frac{3}{8}$	3·78	1·87	124·0
$\frac{1}{3}$	3·48	1·66	118·0
$\frac{1}{4}$	2·97	1·25	108·1
$\frac{1}{5}$	2·60	1·00	102·1
$\frac{1}{6}$	2·58	·83	97·0
$\frac{1}{7}$	2·09	·71	91·9
$\frac{1}{8}$	1·91	·62	86·8
$\frac{1}{9}$	1·77	·55	81·8
$\frac{1}{10}$	1·65	·50	77·0
$\frac{1}{11}$	1·53	·45	72·5
$\frac{1}{12}$	1·44	·41	68·3
$\frac{1}{13}$	1·36	·38	64·3
$\frac{1}{14}$	1·30	·35	60·6
$\frac{1}{15}$	1·23	·33	57·2
$\frac{1}{16}$	1·17	·31	53·9
$\frac{1}{17}$	1·12	·29	50·7
$\frac{1}{18}$	1·07	·27	47·6
$\frac{1}{19}$	1·03	·26	45·1
$\frac{1}{20}$	1·00	·25	43·1
$\frac{1}{25}$	·84	·20	37·7
$\frac{1}{30}$	·73	·16	35·0

10 lb. absolute. Specific vol., 2,358. Temp., 193·9° F.

10 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
$\frac{3}{4}$	9·64	7·50	179·9
$\frac{5}{8}$	9·12	6·25	172·1
$\frac{1}{2}$	8·46	5·00	162·3
$\frac{3}{8}$	7·39	3·75	150·2
$\frac{1}{3}$	6·95	3·33	143·1
$\frac{1}{4}$	5·95	2·50	133·9
$\frac{1}{5}$	5·21	2·00	126·3
$\frac{1}{6}$	4·64	1·66	118·0
$\frac{1}{7}$	4·19	1·42	112·1
$\frac{1}{8}$	3·83	1·25	108·1
$\frac{1}{9}$	3·55	1·11	104·7
$\frac{1}{10}$	3·33	1·00	102·1
$\frac{1}{11}$	3·08	·909	99·8
$\frac{1}{12}$	2·89	·833	97·0
$\frac{1}{13}$	2·72	·769	94·5
$\frac{1}{14}$	2·57	·714	92·0
$\frac{1}{15}$	2·45	·666	89·2
$\frac{1}{16}$	2·35	·625	86·8
$\frac{1}{17}$	2·25	·588	84·0
$\frac{1}{18}$	2·15	·555	81·8
$\frac{1}{19}$	2·06	·526	79·7
$\frac{1}{20}$	1·99	·500	75·5
$\frac{1}{25}$	1·68	·400	65·0
$\frac{1}{30}$	1·46	·333	57·2

15 lb. absolute. Specific vol., 1,610. Temp., 213·4° F.

15 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
$\frac{3}{4}$	14·49	11·25	198·9
$\frac{5}{8}$	13·78	9·44	190·7
$\frac{1}{2}$	12·69	7·50	179·9
$\frac{3}{8}$	11·18	5·66	167·7
$\frac{1}{3}$	10·45	5·00	162·3
$\frac{1}{4}$	8·92	3·75	150·2
$\frac{1}{5}$	7·82	3·00	141·6
$\frac{1}{6}$	6·97	2·50	133·9
$\frac{1}{7}$	6·29	2·14	128·4
$\frac{1}{8}$	5·75	1·87	123·1
$\frac{1}{9}$	5·33	1·66	118·0
$\frac{1}{10}$	4·95	1·50	114·2
$\frac{1}{11}$	4·61	1·36	110·9
$\frac{1}{12}$	4·33	1·25	108·1
$\frac{1}{13}$	4·07	1·15	105·7
$\frac{1}{14}$	3·86	1·07	103·8
$\frac{1}{15}$	3·68	1·00	102·1
$\frac{1}{16}$	3·52	·93	99·6
$\frac{1}{17}$	3·37	·88	98·9
$\frac{1}{18}$	3·23	·83	97·0
$\frac{1}{19}$	3·10	·79	95·0
$\frac{1}{20}$	2·96	·75	93·4
$\frac{1}{25}$	2·52	·60	85·6
$\frac{1}{30}$	2·20	·50	77·0

TABLE II—(*continued*)

EXPANSION OF STEAM

20 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	25 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	30 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
20 lb. absolute. Specific vol., 1,229. Temp., 228° F.					25 lb. absolute. Specific vol., 996. Temp., 240·1° F.					30 lb. absolute. Specific vol., 838. Temp., 250·4° F.				
	3/4	19·3	15·0	213·2		3/4	23·9	18·6	224·1		3/4	28·9	22·5	234·3
	5/8	18·2	12·5	203·9		5/8	22·7	15·6	215·1		5/8	27·3	18·7	224·4
	1/2	16·9	10·0	193·5		1/2	21·1	12·5	203·9		1/2	25·3	15·0	213·2
	3/8	14·7	7·50	179·9		3/8	18·4	9·36	190·3		3/8	22·0	11·2	198·7
	1/3	13·9	6·66	174·7		1/3	17·4	8·33	184·7		1/3	20·9	10·0	193·5
	1/4	11·9	5·00	162·3		1/4	14·8	6·25	172·1		1/4	17·8	7·50	179·9
	1/5	10·43	4·00	153·1		1/5	13·0	5·00	162·3		1/5	15·6	6·00	170·5
	1/6	9·28	3·33	145·3		1/6	11·6	4·16	154·5		1/6	13·9	5·00	162·3
	1/7	8·39	2·85	139·3		1/7	10·4	3·55	147·9		1/7	12·5	4·28	155·6
	1/8	7·67	2·50	133·9		1/8	9·57	3·12	142·9		1/8	11·5	3·75	150·2
	1/9	7·11	2·22	129·6		1/9	8·86	2·77	138·0		1/9	10·6	3·33	145·3
	1/10	6·66	2·00	126·3		1/10	8·25	2·50	133·9		1/10	9·90	3·00	141·6
	1/11	6·13	1·81	121·7		1/11	7·66	2·26	130·2		1/11	9·22	2·72	137·3
	1/12	5·76	1·66	118·0		1/12	7·21	2·08	127·5		1/12	8·67	2·50	133·9
	1/13	5·41	1·53	114·8		1/13	6·79	1·92	124·3		1/13	8·14	2·30	130·8
	1/14	5·12	1·42	112·2		1/14	6·42	1·78	120·9		1/14	7·72	2·14	128·4
	1/15	4·89	1·33	110·0		1/15	6·10	1·66	118·1		1/15	7·36	2·00	126·3
	1/16	4·70	1·25	108·1		1/16	5·86	1·56	115·6		1/16	7·03	1·87	123·1
	1/17	4·48	1·17	106·2		1/17	5·63	1·47	113·4		1/17	6·77	1·77	120·7
	1/18	4·29	1·11	104·7		1/18	5·34	1·38	111·2		1/18	6·42	1·66	118·0
	1/19	4·12	1·05	103·3		1/19	5·14	1·31	109·6		1/19	6·13	1·56	115·6
	1/20	3·99	1·00	102·1		1/20	4·98	1·25	108·1		1/20	5·98	1·50	114·2
	1/25	3·36	·80	95·5		1/25	4·21	1·00	102·1		1/25	5·05	1·20	106·9
	1/30	2·93	·66	89·2		1/30	3·65	·83	97·0		1/30	4·40	1·00	102·1

TABLE II—(*continued*)

Expansion of Steam

35 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	40 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	45 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
35 lb. absolute. Specific vol., 726. Temp., 259·8° F.				40 lb. absolute. Specific vol., 640. Temp., 267·9° F.				45 lb. absolute. Specific vol., 572. Temp., 274·9° F.			
3/4	33·8	26·1	243·0	3/4	38·5	30·0	250·4	3/4	43·4	33·7	257·0
5/8	32·1	21·8	232·3	5/8	36·7	25·0	240·1	5/8	41·3	28·0	246·4
1/2	29·6	17·5	221·1	1/2	33·8	20·0	228·0	1/2	38·1	22·5	236·7
3/8	25·9	13·1	206·2	3/8	29·6	15·0	213·2	3/8	33·3	16·8	218·9
1/3	25·5	11·6	198·7	1/3	27·8	13·3	207·0	1/3	33·0	15·0	213·2
1/4	20·8	8·75	187·1	1/4	23·8	10·0	193·5	1/4	26·8	11·2	198·8
1/5	18·2	7·00	177·0	1/5	20·8	8·0	182·9	1/5	23·4	9·00	188·5
1/6	16·2	5·82	167·6	1/6	18·5	6·6	174·4	1/6	20·9	7·50	179·9
1/7	14·7	5·00	162·3	1/7	16·8	5·5	167·2	1/7	18·8	6·42	173·2
1/8	13·4	4·72	159·3	1/8	15·4	5·0	162·3	1/8	17·3	5·62	167·3
1/9	12·3	3·88	151·7	1/9	14·1	4·44	157·1	1/9	15·9	5·00	162·3
1/10	11·5	3·50	147·2	1/10	13·2	4·00	153·1	1/10	14·8	4·50	157·6
1/11	10·7	3·16	143·4	1/11	12·3	3·63	148·8	1/11	13·9	4·09	153·9
1/12	10·1	2·91	141·0	1/12	11·5	3·33	145·3	1/12	13·3	3·75	150·2
1/13	9·5	2·53	134·0	1/13	10·9	3·06	142·3	1/13	12·3	3·46	146·8
1/14	9·0	2·5	133·0	1/14	10·3	2·92	140·3	1/14	11·6	3·21	144·0
1/15	8·6	2·33	131·3	1/15	9·8	2·66	137·3	1/15	11·1	3·00	141·6
1/16	8·2	2·18	129·0	1/16	9·4	2·50	134·9	1/16	10·5	2·81	138·6
1/17	7·8	2·06	127·2	1/17	8·9	2·35	131·6	1/17	10·0	2·64	136·0
1/18	7·5	1·94	124·8	1/18	8·5	2·22	129·6	1/18	9·7	2·50	133·9
1/19	7·2	1·84	122·4	1/19	8·2	2·10	127·8	1/19	9·2	2·36	131·7
1/20	6·9	1·75	120·2	1/20	7·9	2·00	126·3	1/20	8·9	2·25	130·1
1/25	5·9	1·40	111·9	1/25	6·7	1·60	110·6	1/25	7·5	1·80	121·4
1/30	5·1	1·16	160·0	1/30	5·8	1·33	106·7	1/30	6·6	1·50	114·2

TABLE II—(*continued*)

Expansion of Steam

50 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	55 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	60 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
50 lb. absolute. Specific vol., 518. Temp., 281·4° F.	3/4	48·2	37·50	263·0	55 lb. absolute. Specific vol., 474. Temp., 287·4° F.	3/4	53·1	41·2	269·0	60 lb. absolute. Specific vol., 437. Temp., 292·9° F.	3/4	57·8	45·0	274·4
	5/8	45·9	31·20	252·4		5/8	50·4	34·3	258·1		5/8	55·1	37·5	263·5
	1/2	42·3	25·00	240·1		1/2	46·5	27·5	245·4		1/2	50·7	30·0	250·4
	3/8	37·0	18·70	224·4		3/8	40·4	20·6	229·5		3/8	44·5	22·5	234·3
	1/3	36·5	16·60	218·3		1/3	38·6	18·3	223·3		1/3	41·8	20·0	228·0
	1/4	29·8	12·50	203·9		1/4	31·6	13·7	208·5		1/4	35·7	15·0	213·2
	1/5	26·0	10·00	193·5		1/5	28·7	11·0	197·9		1/5	31·2	12·0	202·0
	1/6	23·2	8·33	184·7		1/6	25·5	9·16	189·3		1/6	27·9	10·0	193·5
	1/7	20·9	7·14	177·8		1/7	23·0	7·85	182·0		1/7	25·6	8·57	186·0
	1/8	19·2	6·25	172·1		1/8	21·1	6·87	176·1		1/8	23·1	7·50	179·9
	1/9	17·5	5·55	166·8		1/9	19·2	6·11	171·2		1/9	21·2	6·60	174·7
	1/10	16·5	5·00	162·3		1/10	18·1	5·50	166·4		1/10	19·8	6·00	170·5
	1/11	15·4	4·54	158·0		1/11	16·9	5·00	162·3		1/11	18·4	5·45	166·0
	1/12	14·4	4·16	152·5		1/12	15·9	4·58	155·7		1/12	17·4	5·00	162·3
	1/13	13·5	3·84	151·2		1/13	15·0	4·23	155·2		1/13	16·1	4·61	158·7
	1/14	12·9	3·57	148·1		1/14	14·2	3·92	152·1		1/14	15·5	4·28	155·6
	1/15	12·2	3·33	145·4		1/15	13·5	3·66	149·1		1/15	14·8	4·00	153·1
	1/16	11·7	3·12	142·9		1/16	12·8	3·43	146·5		1/16	14·1	3·75	150·2
	1/17	11·2	2·94	140·6		1/17	12·3	3·23	144·2		1/17	13·4	3·52	147·4
	1/18	10·7	2·77	138·0		1/18	11·8	3·05	141·8		1/18	12·8	3·33	145·4
	1/19	10·3	2·63	135·9		1/19	11·3	2·89	139·9		1/19	12·4	3·15	143·3
	1/20	9·97	2·50	133·9		1/20	9·97	2·75	137·7		1/20	11·9	3·00	141·6
	1/25	8·42	2·00	126·3		1/25	9·26	2·20	129·3		1/25	10·1	2·40	132·4
	1/30	7·30	1·66	118·0		1/30	8·06	1·83	122·4		1/30	8·8	2·00	126·3

TABLE II—(*continued*)

Expansion of Steam

65 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	70 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	75 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
65 lb. absolute. Specific vol., 405. Temp., 298° F.				70 lb. absolute. Specific vol., 378. Temp., 302·8° F.				75 lb. absolute. Specific vol., 353. Temp., 307·5° F.			
3/4	62·4	48·75	279·3	3/4	67·4	52·5	284·1	3/4	72·0	56·1	288·8
5/8	59·7	40·60	268·2	5/8	64·3	44·4	273·7	5/8	68·4	46·8	276·8
1/2	54·9	32·50	255·0	1/2	59·2	35·0	259·3	1/2	63·3	37·5	263·4
3/8	48·3	24·30	238·6	3/8	52·4	26·6	243·5	3/8	55·3	28·1	246·6
1/3	45·1	20·60	232·1	1/3	51·3	23·3	236·2	1/3	52·2	25·0	240·1
1/4	38·6	16·20	217·1	1/4	41·7	17·5	221·0	1/4	44·5	18·7	224·4
1/5	33·9	13·00	205·9	1/5	36·4	14·0	209·7	1/5	39·0	15·0	213·2
1/6	30·2	10·80	197·0	1/6	32·3	11·6	200·3	1/6	34·8	12·5	203·9
1/7	27·2	9·28	189·9	1/7	29·4	10·0	193·8	1/7	31·4	10·7	196·5
1/8	23·9	8·12	183·5	1/8	26·9	8·75	187·1	1/8	28·7	9·37	190·5
1/9	23·0	7·22	178·2	1/9	24·7	7·77	181·5	1/9	26·5	8·33	184·7
1/10	21·4	6·50	173·7	1/10	23·1	7·00	177·0	1/10	24·7	7·50	179·9
1/11	20·0	5·90	169·6	1/11	21·5	6·36	172·8	1/11	23·0	6·81	175·7
1/12	18·8	5·41	165·5	1/12	20·2	5·83	169·1	1/12	21·7	6·25	172·1
1/13	17·8	5·00	162·3	1/13	19·1	5·38	165·4	1/13	20·5	5·76	168·5
1/14	16·8	4·64	158·9	1/14	18·1	5·00	162·3	1/14	19·4	5·37	165·3
1/15	16·2	4·33	156·0	1/15	17·2	4·66	159·1	1/15	18·8	5·00	162·3
1/16	15·2	4·06	153·6	1/16	16·4	4·37	156·5	1/16	17·5	4·68	159·3
1/17	14·5	3·82	151·0	1/17	15·6	4·11	154·3	1/17	16·8	4·41	156·7
1/18	14·0	3·61	148·4	1/18	14·9	3·88	151·6	1/18	16·1	4·16	154·5
1/19	13·4	3·42	146·3	1/19	14·4	3·68	149·2	1/19	15·7	3·95	152·5
1/20	12·9	3·25	144·4	1/20	14·0	3·50	147·3	1/20	14·9	3·75	150·2
1/25	10·9	2·70	137·0	1/25	11·7	2·80	138·5	1/25	12·6	3·00	141·6
1/30	9·5	2·15	128·5	1/30	10·2	2·33	131·3	1/30	11·0	2·50	133·9

TABLE II—(*continued*)

Expansion of Steam

80 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	90 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	100 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
80 lb. absolute. Specific vol., 333. Temp., 312° F.	$\frac{3}{4}$	77·1	60·0	292·7	90lb. absolute. Specific vol., 298. Temp., 320·2° F.	$\frac{3}{4}$	86·7	67·5	300·5	100 lb. absolute. Specific vol., 270. Temp., 327·5° F.	$\frac{3}{4}$	96·3	75·0	307·5
	$\frac{5}{8}$	73·5	50·0	281·0		$\frac{5}{8}$	82·6	56·0	288·3		$\frac{5}{8}$	91·8	62·5	295·4
	$\frac{1}{2}$	67·7	40·0	267·3		$\frac{1}{2}$	76·1	45·0	274·4		$\frac{1}{2}$	84·6	50·0	281·0
	$\frac{3}{8}$	59·3	30·0	250·4		$\frac{3}{8}$	66·7	33·6	256·9		$\frac{3}{8}$	74·1	37·5	265·0
	$\frac{1}{3}$	58·5	26·6	243·5		$\frac{1}{3}$	66·0	30·0	250·4		$\frac{1}{3}$	73·3	33·3	257·4
	$\frac{1}{4}$	47·5	20·0	228·0		$\frac{1}{4}$	53·6	22·5	234·3		$\frac{1}{4}$	59·6	25·0	240·1
	$\frac{1}{5}$	41·5	16·0	216·5		$\frac{1}{5}$	46·8	18·0	222·5		$\frac{1}{5}$	52·1	20·0	228·0
	$\frac{1}{6}$	37·1	13·3	207·0		$\frac{1}{6}$	41·8	15·0	213·2		$\frac{1}{6}$	46·3	16·6	218·3
	$\frac{1}{7}$	33·5	11·4	199·5		$\frac{1}{7}$	37·6	12·8	205·1		$\frac{1}{7}$	41·7	14·2	210·4
	$\frac{1}{8}$	30·8	10·0	193·5		$\frac{1}{8}$	34·6	11·2	198·3		$\frac{1}{8}$	38·4	12·5	203·9
	$\frac{1}{9}$	28·3	8·88	187·8		$\frac{1}{9}$	31·9	10·0	193·5		$\frac{1}{9}$	35·4	11·1	198·3
	$\frac{1}{10}$	26·4	8·00	182·9		$\frac{1}{10}$	29·7	9·0	188·5		$\frac{1}{10}$	33·0	10·0	193·5
	$\frac{1}{11}$	24·6	7·27	178·5		$\frac{1}{11}$	26·7	8·18	183·9		$\frac{1}{11}$	30·8	9·09	188·9
	$\frac{1}{12}$	23·1	6·66	174·7		$\frac{1}{12}$	26·1	7·50	179·9		$\frac{1}{12}$	29·0	8·33	184·7
	$\frac{1}{13}$	21·8	6·15	171·4		$\frac{1}{13}$	24·6	6·92	176·4		$\frac{1}{13}$	27·3	7·69	181·0
	$\frac{1}{14}$	20·7	5·71	168·1		$\frac{1}{14}$	23·3	6·42	173·2		$\frac{1}{14}$	25·9	7·14	177·8
	$\frac{1}{15}$	20·0	5·33	165·0		$\frac{1}{15}$	22·5	6·00	170·5		$\frac{1}{15}$	25·0	6·66	174·7
	$\frac{1}{16}$	18·8	5·00	162·3		$\frac{1}{16}$	21·1	5·62	167·3		$\frac{1}{16}$	23·5	6·25	172·1
	$\frac{1}{17}$	17·9	4·71	159·6		$\frac{1}{17}$	20·2	5·29	164·6		$\frac{1}{17}$	22·5	5·88	169·5
	$\frac{1}{18}$	17·2	4·44	157·1		$\frac{1}{18}$	19·4	5·00	162·3		$\frac{1}{18}$	21·5	5·55	166·8
	$\frac{1}{19}$	16·5	4·21	155·0		$\frac{1}{19}$	18·6	4·73	157·8		$\frac{1}{19}$	20·72	5·26	164·4
	$\frac{1}{20}$	15·9	4·00	153·1		$\frac{1}{20}$	18·0	4·50	155·7		$\frac{1}{20}$	19·9	5·00	162·3
	$\frac{1}{25}$	13·4	3·20	143·9		$\frac{1}{25}$	15·1	3·60	148·6		$\frac{1}{25}$	16·8	4·00	153·1
	$\frac{1}{30}$	11·7	2·66	136·3		$\frac{1}{30}$	13·2	3·00	141·6		$\frac{1}{30}$	14·6	3·33	145·3

TABLE II—(*continued*)

Expansion of Steam

110 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	120 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terimnal Temp. at same.	130 lb.	Average Pressure when cut off at		Terimnal Pressure at same.	Terminal Temp. at same.
110 lb. absolute. Specific vol., 247. Temp., 334·5° F.	$\frac{3}{4}$	106·0	82·5	314·2	120 lb. absolute. Specific vol., 227. Temp., 340·8° F.	$\frac{3}{4}$	115·2	90·0	320·2	130 lb. absolute. Specific vol., 211·5. Temp., 346·8° F.	$\frac{3}{4}$	125·4	97·5	326·2
	$\frac{5}{8}$	101·0	68·7	301·8		$\frac{5}{8}$	110·2	75·0	307·5		$\frac{5}{8}$	119·1	81·2	312·9
	$\frac{1}{2}$	93·1	55·0	287·1		$\frac{1}{2}$	101·5	60·0	292·7		$\frac{1}{2}$	110·0	65·0	298·0
	$\frac{3}{8}$	81·5	41·2	269·0		$\frac{3}{8}$	89·4	45·0	274·4		$\frac{3}{8}$	96·2	48·7	278·9
	$\frac{1}{3}$	80·0	36·6	261·9		$\frac{1}{3}$	83·6	40·0	267·3		$\frac{1}{3}$	90·4	43·3	272·0
	$\frac{1}{4}$	65·6	27·5	245·4		$\frac{1}{4}$	71·5	30·0	250·4		$\frac{1}{4}$	77·5	32·5	255·0
	$\frac{1}{5}$	58·2	22·0	233·1		$\frac{1}{5}$	62·4	24·0	237·8		$\frac{1}{5}$	67·8	26·0	242·3
	$\frac{1}{6}$	51·1	18·3	223·3		$\frac{1}{6}$	55·8	20·0	228·0		$\frac{1}{6}$	59·4	21·6	232·1
	$\frac{1}{7}$	46·1	15·7	215·5		$\frac{1}{7}$	50·2	17·1	219·8		$\frac{1}{7}$	53·3	18·5	223·9
	$\frac{1}{8}$	42·5	13·7	208·5		$\frac{1}{8}$	46·1	15·0	213·2		$\frac{1}{8}$	50·0	16·2	217·1
	$\frac{1}{9}$	38·9	12·2	202·7		$\frac{1}{9}$	43·4	13·3	207·0		$\frac{1}{9}$	45·9	14·4	211·1
	$\frac{1}{10}$	36·3	11·0	197·9		$\frac{1}{10}$	39·6	12·0	202·0		$\frac{1}{10}$	42·9	13·0	205·9
	$\frac{1}{11}$	33·3	10·0	193·5		$\frac{1}{11}$	36·9	10·9	197·4		$\frac{1}{11}$	40·0	11·8	201·1
	$\frac{1}{12}$	31·8	9·16	189·3		$\frac{1}{12}$	34·8	10·0	193·5		$\frac{1}{12}$	37·5	10·8	197·0
	$\frac{1}{13}$	30·1	8·46	185·4		$\frac{1}{13}$	32·8	9·23	190·0		$\frac{1}{13}$	35·6	10·0	193·5
	$\frac{1}{14}$	28·4	7·85	182·0		$\frac{1}{14}$	31·1	8·57	186·0		$\frac{1}{14}$	33·6	9·28	189·9
	$\frac{1}{15}$	27·6	7·33	178·9		$\frac{1}{15}$	30·08	8·00	182·9		$\frac{1}{15}$	32·5	8·66	186·5
	$\frac{1}{16}$	25·8	6·87	176·1		$\frac{1}{16}$	28·20	7·50	179·9		$\frac{1}{16}$	30·5	8·12	183·5
	$\frac{1}{17}$	24·7	6·47	173·5		$\frac{1}{17}$	26·93	7·05	177·2		$\frac{1}{17}$	29·1	7·64	180·7
	$\frac{1}{18}$	23·7	6·11	171·2		$\frac{1}{18}$	25·73	6·66	174·7		$\frac{1}{18}$	28·0	7·22	178·2
	$\frac{1}{19}$	22·8	5·79	168·7		$\frac{1}{19}$	24·86	6·31	172·5		$\frac{1}{19}$	26·9	6·84	175·9
	$\frac{1}{20}$	21·8	5·50	166·4		$\frac{1}{20}$	24·00	6·00	170·5		$\frac{1}{20}$	25·9	6·50	173·7
	$\frac{1}{25}$	18·6	4·44	157·1		$\frac{1}{25}$	20·20	4·80	160·4		$\frac{1}{25}$	21·8	5·20	163·9
	$\frac{1}{30}$	16·1	3·66	149·1		$\frac{1}{30}$	17·70	4·00	153·1		$\frac{1}{30}$	19·0	4·33	156·1

TABLE II—(*continued*)

Expansion of Steam

140 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	150 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	160 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
140 lb. absolute. Specific vol., 196·5 Temp., 352·8° F.	$\frac{3}{4}$	134·9	105·0	331·2	150 lb. absolute. Specific vol., 185. Temp., 358·2° F.	$\frac{3}{4}$	144·7	112·5	336·4	160 lb. absolute. Specific vol., 175. Temp., 363·2° F.	$\frac{3}{4}$	153·6	120·0	341·2
	$\frac{5}{8}$	128·6	87·5	318·2		$\frac{5}{8}$	137·8	93·7	323·1		$\frac{5}{8}$	147·0	100·0	328·2
	$\frac{1}{2}$	118·5	70·0	303·0		$\frac{1}{2}$	126·4	75·0	307·5		$\frac{1}{2}$	135·4	80·0	312·0
	$\frac{3}{8}$	103·8	52·5	284·1		$\frac{3}{8}$	111·2	56·2	289·0		$\frac{3}{8}$	118·2	60·0	292·7
	$\frac{1}{3}$	97·4	46·6	276·5		$\frac{1}{3}$	104·5	50·0	281·0		$\frac{1}{3}$	115·5	53·3	285·0
	$\frac{1}{4}$	83·3	35·0	259·3		$\frac{1}{4}$	89·4	37·5	263·4		$\frac{1}{4}$	95·4	40·0	267·3
	$\frac{1}{5}$	72·8	28·0	246·4		$\frac{1}{5}$	78·2	30·0	250·4		$\frac{1}{5}$	83·2	32·0	254·1
	$\frac{1}{6}$	65·0	23·3	236·1		$\frac{1}{6}$	69·7	25·0	240·1		$\frac{1}{6}$	74·2	26·6	243·5
	$\frac{1}{7}$	58·0	20·0	228·0		$\frac{1}{7}$	62·9	21·4	231·4		$\frac{1}{7}$	67·0	22·8	234·0
	$\frac{1}{8}$	53·8	17·5	220·0		$\frac{1}{8}$	57·7	18·7	224·5		$\frac{1}{8}$	61·5	20·0	228·0
	$\frac{1}{9}$	49·4	15·5	214·8		$\frac{1}{9}$	52·9	16·6	218·3		$\frac{1}{9}$	56·4	17·7	221·6
	$\frac{1}{10}$	46·2	14·0	209·7		$\frac{1}{10}$	49·5	15·0	213·2		$\frac{1}{10}$	52·8	16·0	216·5
	$\frac{1}{11}$	43·0	12·7	204·7		$\frac{1}{11}$	46·1	13·6	208·7		$\frac{1}{11}$	49·1	14·5	211·4
	$\frac{1}{12}$	40·3	11·6	200·2		$\frac{1}{12}$	43·5	12·5	203·9		$\frac{1}{12}$	46·2	13·3	207·0
	$\frac{1}{13}$	38·8	10·9	197·4		$\frac{1}{13}$	41·2	11·6	200·3		$\frac{1}{13}$	43·7	12·3	203·1
	$\frac{1}{14}$	36·3	10·0	193·5		$\frac{1}{14}$	38·8	10·7	196·5		$\frac{1}{14}$	41·4	11·4	199·5
	$\frac{1}{15}$	35·0	9·33	190·1		$\frac{1}{15}$	37·6	10·0	193·5		$\frac{1}{15}$	39·8	10·6	196·1
	$\frac{1}{16}$	33·9	8·75	187·1		$\frac{1}{16}$	36·3	9·37	190·3		$\frac{1}{16}$	37·6	10·0	193·5
	$\frac{1}{17}$	31·4	8·23	184·1		$\frac{1}{17}$	33·6	8·82	187·4		$\frac{1}{17}$	35·9	9·41	190·5
	$\frac{1}{18}$	30·1	7·77	181·5		$\frac{1}{18}$	32·3	8·33	184·7		$\frac{1}{18}$	34·0	8·88	187·8
	$\frac{1}{19}$	28·9	7·36	179·1		$\frac{1}{19}$	31·0	7·89	182·2		$\frac{1}{19}$	33·1	8·42	185·2
	$\frac{1}{20}$	27·6	7·00	177·0		$\frac{1}{20}$	29·9	7·50	179·9		$\frac{1}{20}$	32·0	8·00	182·9
	$\frac{1}{25}$	23·5	5·6	167·2		$\frac{1}{25}$	25·2	6·00	170·5		$\frac{1}{25}$	26·9	6·40	173·1
	$\frac{1}{30}$	20·4	4·66	159·1		$\frac{1}{30}$	22·0	5·00	162·3		$\frac{1}{30}$	23·4	5·33	165·0

TABLE II—(*continued*)

EXPANSION OF STEAM

170 lb. absolute. Specific vol., 165. Temp., 368·2° F.

180 lb. absolute. Specific vol., 156. Temp., 373·2° F.

200 lb. absolute. Specific vol., 141. Temp., 381·6° F.

170 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	180 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	200 lb. Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
3/4	164·2	127·5	345·7	3/4	173·5	135·0	350·2	3/4	192·7	150·0	358·5
5/8	162·3	111·2	335·5	5/8	164·6	112·5	336·4	5/8	183·7	125·0	344·2
1/2	142·9	85·0	316·1	1/2	152·3	90·0	320·2	1/2	169·3	100·0	328·2
3/8	131·6	66·7	299·7	3/8	132·9	67·5	300·5	3/8	148·3	75·0	307·5
1/3	118·2	56·6	289·4	1/3	125·4	60·0	292·7	1/3	139·2	66·6	299·6
1/4	101·1	42·5	270·9	1/4	107·3	45·0	274·4	1/4	119·3	50·0	281·0
1/5	88·7	34·0	257·6	1/5	88·7	34·0	257·6	1/5	104·3	40·0	267·3
1/6	78·9	28·3	247·0	1/6	83·7	30·0	250·4	1/6	92·9	33·3	256·4
1/7	71·1	24·2	238·2	1/7	71·7	25·7	241·6	1/7	83·7	28·5	247·4
1/8	65·2	21·2	231·1	1/8	69·2	22·5	234·3	1/8	76·9	25·0	240·1
1/9	59·9	18·8	224·6	1/9	64·0	20·0	228·0	1/9	71·0	22·2	233·5
1/10	56·1	17·0	219·6	1/10	59·4	18·0	222·6	1/10	66·0	20·0	228·0
1/11	52·2	15·4	214·7	1/11	55·2	16·3	217·4	1/11	61·3	18·1	222·8
1/12	49·2	14·1	210·0	1/12	52·0	15·0	213·2	1/12	57·8	16·6	218·7
1/13	46·2	13·0	205·9	1/13	48·9	13·8	208·9	1/13	54·1	15·3	214·1
1/14	44·0	12·1	202·3	1/14	46·4	12·8	205·1	1/14	51·5	14·2	210·4
1/15	42·2	11·3	199·1	1/15	44·1	12·0	202·0	1/15	49·0	13·3	207·3
1/16	39·3	10·6	196·1	1/16	42·3	11·2	198·7	1/16	47·0	12·5	203·9
1/17	38·2	10·0	193·5	1/17	40·4	10·5	195·7	1/17	44·6	11·7	200·7
1/18	36·6	9·44	190·8	1/18	38·8	10·0	193·5	1/18	43·0	11·1	198·3
1/19	35·2	8·94	188·1	1/19	37·3	9·47	190·8	1/19	41·3	10·5	195·7
1/20	33·9	8·50	185·7	1/20	36·0	9·00	188·5	1/20	39·9	10·0	193·5
1/25	28·6	6·80	175·7	1/25	30·3	7·20	178·0	1/25	33·6	8·00	182·9
1/30	24·3	5·66	165·7	1/30	26·4	6·00	170·5	1/30	29·3	6·66	174·7

TABLE II—(*continued*)

EXPANSION OF STEAM

220 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	240 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.	260 lb.	Average Pressure when cut off at		Terminal Pressure at same.	Terminal Temp. at same.
220 lb. absolute. Specific vol., 129. Temp., 389·6° F.	$\frac{3}{4}$	211·2	165·0	365·9	240 lb., absolute. Specific vol., 119. Temp., 397·6° F.	$\frac{3}{4}$	231·8	180·0	373·0	260 lb. absolute. Specific vol., 111. Temp., 404·6° F.	$\frac{3}{4}$	251·1	195·0	379·8
	$\frac{5}{8}$	200·7	137·5	351·7		$\frac{5}{8}$	219·0	150·0	358·5		$\frac{5}{8}$	227·2	162·5	365·0
	$\frac{1}{2}$	186·1	110·0	334·7		$\frac{1}{2}$	202·8	120·0	341·2		$\frac{1}{2}$	219·7	130·0	347·2
	$\frac{3}{8}$	162·7	82·5	314·1		$\frac{3}{8}$	177·5	90·0	320·2		$\frac{3}{8}$	192·3	97·5	326·2
	$\frac{1}{3}$	153·1	73·3	305·9		$\frac{1}{3}$	167·2	80·0	312·0		$\frac{1}{3}$	180·9	86·6	317·4
	$\frac{1}{4}$	130·9	55·0	287·1		$\frac{1}{4}$	142·8	60·0	292·7		$\frac{1}{4}$	154·7	65·0	298·0
	$\frac{1}{5}$	114·7	44·0	273·0		$\frac{1}{5}$	125·0	48·0	278·4		$\frac{1}{5}$	135·6	52·0	283·5
	$\frac{1}{6}$	101·2	36·6	261·9		$\frac{1}{6}$	111·6	40·0	267·3		$\frac{1}{6}$	120·8	43·3	272·0
	$\frac{1}{7}$	96·1	32·7	255·3		$\frac{1}{7}$	100·5	34·2	257·9		$\frac{1}{7}$	104·5	35·5	261·1
	$\frac{1}{8}$	82·5	27·5	245·4		$\frac{1}{8}$	92·1	30·0	250·4		$\frac{1}{8}$	99·7	32·5	255·0
	$\frac{1}{9}$	78·0	24·4	238·6		$\frac{1}{9}$	85·3	26·6	242·4		$\frac{1}{9}$	92·4	28·8	247·2
	$\frac{1}{10}$	72·6	22·0	233·1		$\frac{1}{10}$	79·2	24·0	237·8		$\frac{1}{10}$	85·8	26·0	242·3
	$\frac{1}{11}$	67·8	20·0	228·0		$\frac{1}{11}$	73·2	21·6	231·9		$\frac{1}{11}$	80·1	23·6	236·8
	$\frac{1}{12}$	63·5	18·3	223·3		$\frac{1}{12}$	69·4	20·0	228·0		$\frac{1}{12}$	75·1	21·6	232·1
	$\frac{1}{13}$	59·8	16·9	219·2		$\frac{1}{13}$	65·1	18·4	223·6		$\frac{1}{13}$	70·8	20·0	228·0
	$\frac{1}{14}$	56·1	15·5	214·8		$\frac{1}{14}$	61·8	17·1	219·9		$\frac{1}{14}$	66·9	18·5	223·9
	$\frac{1}{15}$	53·9	14·6	211·8		$\frac{1}{15}$	58·8	16·0	216·5		$\frac{1}{15}$	63·7	17·3	220·4
	$\frac{1}{16}$	51·7	13·7	208·5		$\frac{1}{16}$	56·4	15·0	213·2		$\frac{1}{16}$	61·0	16·2	217·1
	$\frac{1}{17}$	49·6	12·9	205·5		$\frac{1}{17}$	54·0	14·1	210·0		$\frac{1}{17}$	58·5	15·3	214·8
	$\frac{1}{18}$	47·6	12·2	202·7		$\frac{1}{18}$	51·5	13·3	207·0		$\frac{1}{18}$	55·8	14·4	211·1
	$\frac{1}{19}$	45·6	11·5	200·4		$\frac{1}{19}$	49·5	12·6	204·3		$\frac{1}{19}$	53·7	13·6	208·1
	$\frac{1}{20}$	43·8	11·0	197·9		$\frac{1}{20}$	47·8	12·0	202·0		$\frac{1}{20}$	51·8	13·0	205·9
	$\frac{1}{25}$	35·4	8·88	187·9		$\frac{1}{25}$	40·4	9·6	191·5		$\frac{1}{25}$	43·7	10·4	195·2
	$\frac{1}{30}$	32·2	7·33	178·9		$\frac{1}{30}$	35·2	8·0	182·9		$\frac{1}{30}$	38·1	8·66	186·5

TABLE II—(*continued*)

Expansion of Steam

280 lb. absolute. Specific vol., 103. Temp., 410·6° F.

280 lb.	Average Pressure when cut off at	Terminal Pressure at same.	Terminal Temp. at same.
3/4	257·8	210·0	386·0
5/8	255·5	175·0	370·7
1/2	236·6	140·0	353·2
3/8	207·1	105·0	331·2
1/3	195·0	93·3	322·8
1/4	166·6	70·0	303·0
1/5	146·0	56·0	288·3
1/6	130·0	46·6	276·5
1/7	117·6	40·0	267·3
1/8	107·4	35·0	259·3
1/9	99·2	31·1	252·1
1/10	92·4	28·0	246·4
1/11	86·4	25·5	241·2
1/12	80·8	23·3	236·1
1/13	74·3	21·0	230·6
1/14	72·2	20·0	228·0
1/15	69·1	18·8	224·5
1/16	65·8	17·5	221·0
1/17	63·0	16·4	217·7
1/18	60·1	15·5	214·8
1/19	57·8	14·7	212·1
1/20	55·8	14·0	209·7
1/25	47·1	11·2	108·7
1/30	41·0	9·3	190·0

300 lb. absolute. Specific vol., 96. Temp., 416·6° F.

300 lb.	Average Pressure when cut off at	Terminal Pressure at same.	Terminal Temp. at same.
3/4	289·8	225·0	391·6
5/8	273·7	187·5	376·4
1/2	253·5	150·0	358·5
3/8	221·9	112·5	336·4
1/3	209·0	100·0	328·2
1/4	171·5	75·0	307·5
1/5	156·5	60·0	292·7
1/6	139·5	50·0	281·0
1/7	125·9	42·8	271·3
1/8	115·1	37·5	263·2
1/9	106·6	33·3	256·4
1/10	99·0	30·0	250·4
1/11	92·4	27·2	244·8
1/12	86·7	25·0	240·1
1/13	81·4	23·0	235·5
1/14	77·3	21·4	231·6
1/15	73·6	20·0	228·0
1/16	70·4	18·7	224·4
1/17	67·5	17·6	221·3
1/18	64·0	16·6	218·3
1/19	62·1	15·7	215·5
1/20	59·8	15·0	213·2
1/25	50·5	12·0	202·0
1/30	44·0	10·0	193·5

350 lb. absolute. Specific vol., 83. Temp., 431·0° F.

350 lb.	Average Pressure when cut off at	Terminal Pressure at same.	Terminal Temp. at same.
3/4	338·0	262·0	405·7
5/8	319·2	218·7	389·5
1/2	286·1	175·0	370·7
3/8	258·0	131·2	347·9
1/3	243·6	116·6	339·1
1/4	207·2	87·5	318·2
1/5	182·6	70·0	303·0
1/6	162·7	58·3	390·6
1/7	147·0	50·0	281·0
1/8	134·3	43·7	272·5
1/9	124·4	38·8	265·4
1/10	116·5	35·0	259·3
1/11	107·8	31·8	253·5
1/12	101·1	29·1	248·6
1/13	95·2	26·9	244·1
1/14	90·2	25·0	240·1
1/15	85·8	23·3	236·2
1/16	82·2	21·8	232·6
1/17	78·5	20·5	229·3
1/18	75·2	19·2	226·3
1/19	72·3	18·4	223·6
1/20	69·8	17·5	221·0
1/25	59·0	14·0	211·7
1/30	51·3	11·6	200·3

The average pressures have been obtained by the use of constants used in the following manner :—

Constants for finding the average pressure of steam with any point of expansion.

Divide the initial pressure by the number of expansions, and multiply the result by the constant of the same number. This will give the average pressure with that point of cut-off. Thus—

Let initial pressure be 100 lb.

Cut-off at $\frac{1}{5}$, i.e. 5 expansions.

The constant of 5 is 2·609.

Then $\frac{100}{5} \times 2{\cdot}609 = 52{\cdot}18$ lb. average pressure.

The constants are formed by the hyperbolic logarithm of a number +1, and are given in Table III.

TABLE III.

No. of Expansions.	Point of Cut-off.	Constant.	No. of Expansions.	Point of Cut-off.	Constant.	No. of Expansions.	Point of Cut-off.	Constant.
—	1	1·00	9	$\frac{1}{9}$	3·197	20	$\frac{1}{20}$	3·995
—	$\frac{3}{4}$	1·288	10	$\frac{1}{10}$	3·302	21	$\frac{1}{21}$	4·044
—	$\frac{5}{8}$	1·460	11	$\frac{1}{11}$	3·397	22	$\frac{1}{22}$	4·091
2	$\frac{1}{2}$	1·692	12	$\frac{1}{12}$	3·484	23	$\frac{1}{23}$	4·135
—	$\frac{3}{8}$	1·973	13	$\frac{1}{13}$	3·564	24	$\frac{1}{24}$	4·178
3	$\frac{1}{3}$	2·098	14	$\frac{1}{14}$	3·639	25	$\frac{1}{25}$	4·218
4	$\frac{1}{4}$	2·386	15	$\frac{1}{15}$	3·708	26	$\frac{1}{26}$	4·258
5	$\frac{1}{5}$	2·609	16	$\frac{1}{16}$	3·772	27	$\frac{1}{27}$	4·295
6	$\frac{1}{6}$	2·791	17	$\frac{1}{17}$	3·833	28	$\frac{1}{28}$	4·332
7	$\frac{1}{7}$	2·945	18	$\frac{1}{18}$	3·890	29	$\frac{1}{29}$	4·367
8	$\frac{1}{8}$	3·079	19	$\frac{1}{19}$	3·944	30	$\frac{1}{30}$	4·401

In calculating the weight of steam used by an engine it is most convenient to reduce the volume up to the point of cut-off to cubic feet. For this purpose the Table IV will be found very useful as in its third column it gives the area in square feet of all cylinders from one inch diameter up to 96 inches or 8 feet diameter.

Table I in its fourth column gives the specific volume of all pressures up to 1,000 lb. By specific volume is meant the number of cubic feet of steam which are required to make one cubic foot of water. If then the total volume of steam used per minute or per hour, be divided by its specific volume, the result will be the number of cubic feet of water. The number must then be multiplied by the weight in lb. of one cubic foot of water, i.e. 62·4 lb., and the result will be the weight of water in pounds.

TABLE IV

Areas of Circles in Square Inches and Square Feet

Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.
1	·7854	·0054	6¾	35·78	·2484	13¼	137·88	·9569
1⅛	1·227	·0084	7	38·48	·2672	13½	143·13	·9937
1⅜	1·484	·0103	7¼	41·28	·2866	13¾	148·48	1·030
1½	1·767	·0128	7½	44·17	·3067	14	153·93	1·068
1⅝	2·073	·0143	7¾	47·17	·3275	14¼	159·48	1·106
1¾	2·405	·0167	8	50·26	·3490	14½	165·13	1·146
1⅞	2·761	·0191	8¼	53·45	·3711	14¾	170·87	1·186
2	3·141	·0218	8½	56·74	·3940	15	176·71	1·227
2¼	3·976	·0276	8¾	60·13	·4175	15¼	182·65	1·267
2½	4·908	·0340	9	63·61	·4410	15½	188·69	1·309
2¾	5·939	·0412	9¼	67·20	·4666	15¾	194·82	1·352
3	7·068	·0490	9½	70·88	·4922	16	201·06	1·394
3¼	8·295	·0576	9¾	74·66	·5184	16¼	207·39	1·439
3½	9·621	·0668	10	78·54	·5454	16½	213·82	1·484
3¾	11·04	·0766	10¼	82·51	·5732	16¾	220·35	1·529
4	12·56	·0802	10½	86·59	·6013	17	226·98	1·575
4¼	14·18	·0984	10¾	90·76	·6302	17¼	233·70	1·622
4½	15·90	·1104	11	95·03	·6599	17½	240·52	1·670
4¾	17·72	·1230	11¼	99·40	·6902	17¾	247·45	1·717
5	19·63	·1363	11½	103·86	·7208	18	254·46	1·766
5¼	21·64	·1502	11¾	108·43	·7527	18¼	261·58	1·815
5½	23·75	·1647	12	113·09	·7854	18½	268·80	1·866
5¾	25·96	·1802	12¼	117·85	·8180	18¾	276·11	1·917
6	28·27	·1963	12½	122·71	·8520	19	283·52	1·961
6¼	30·67	·2129	12¾	127·67	·8868	19¼	291·03	2·020
6½	33.18	·2304	13	132·73	·9215	19½	298·64	2·070

TABLE IV—(*continued*)

Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.
19¾	300·35	2·127	26½	551·54	3·829	33¼	868·30	6·030
20	314·16	2·181	26¾	562·00	3·902	33½	881·41	6·120
20¼	322·06	2·236	27	572·55	3·975	33¾	894·62	6·212
20½	330·06	2·291	27¼	583·20	4·050	34	907·92	6·235
20¾	339·16	2·347	27½	593·95	4·124	34¼	921·32	6·398
21	346·36	2·404	27¾	604·50	4·200	34½	934·82	6·485
21¼	354·65	2·462	28	615·75	4·275	34¾	948·41	6·586
21½	363·05	2·520	28¼	626·79	4·352	35	962·11	6·682
21¾	371·54	2·579	28½	637·94	4·429	35¼	975·90	6·781
22	380·13	2·639	28¾	649·18	4·507	35½	989·79	6·881
22¼	388·82	2·700	29	660·52	4·586	35¾	1003·78	6·970
22½	397·60	2·761	29¼	671·95	4·665	36	1017·87	7·068
22¾	406·49	2·822	29½	683·49	4·745	36¼	1032·06	7·172
23	415·47	2·884	29¾	695·12	4·827	36½	1046·34	7·264
23¼	424·55	2·947	30	706·86	4·908	36¾	1060·73	7·366
23½	433·73	3·011	30¼	718·69	4·990	37	1075·21	7·466
23¾	443·01	3·076	30½	730·61	5·073	37¼	1089·80	7·568
24	452·39	3·141	30¾	742·64	5·156	37½	1104·46	7·669
24¼	461·86	3·207	31	754·76	5·241	37¾	1119·24	7·772
24½	471·43	3·240	31¼	766·99	5·326	38	1134·16	7·875
24¾	481·10	3·340	31½	779·31	5·410	38¼	1149·08	7·979
25	490·87	3·408	31¾	791·73	5·498	38½	1164·15	8·083
25¼	500·74	3·477	32	804·24	5·585	38¾	1179·32	8·189
25½	510·70	3·546	32¼	816·85	5·672	39	1194·59	8·226
25¾	520·76	3·611	32½	829·57	5·753	39¼	1209·95	8·402
26	530·93	3·686	32¾	842·40	5·856	39½	1225·42	8·509
26¼	541·18	3·757	33	855·30	5·939	39¾	1240·98	8·610

TABLE IV—(*continued*)

Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.
40	1256·64	8·726	46¾	1716·54	11·920	59	2733·97	18·985
40¼	1272·39	8·835	47	1734·94	12·048	59½	2780·51	19·309
40½	1288·25	8·945	47¼	1753·45	12·173	60	2827·44	19·625
40¾	1304·20	9·056	47½	1772·05	12·305	60½	2874·76	19·963
41	1320·25	9·168	47¾	1790·75	12·435	61	2922·47	20·225
41¼	1336·39	9·280	48	1809·56	12·566	61½	2970·57	20·628
41½	1352·65	9·393	48½	1847·45	12·826	62	3019·07	20·965
41¾	1369·00	9·506	49	1885·74	13·157	62½	3067·96	21·305
42	1385·44	9·621	49½	1924·42	13·362	63	3117·25	21·578
42¼	1401·98	9·735	50	1963·50	13·635	63½	3166·92	21·992
42½	1418·62	9·844	50½	2002·96	13·909	64	3216·99	22·331
42¾	1435·36	9·967	51	2042·82	14·186	65	3318·31	23·043
43	1452·20	10·084	51½	2083·07	14·465	66	3421·20	23·758
43¼	1469·13	10·232	52	2123·72	14·744	67	3525·66	24·482
43½	1486·17	10·320	52½	2164·75	15·032	68	3631·68	25·220
43¾	1503·30	10·432	53	2206·18	15·320	69	3739·28	25·966
44	1520·53	10·559	53½	2248·01	15·611	70	3848·46	26·254
44¼	1537·86	10·672	54	2290·22	15·902	71	3959·20	27·494
44½	1555·28	10·800	54½	2332·83	16·200	72	4071·51	28·274
44¾	1572·81	10·922	55	2375·83	16·498	73	4185·39	29·064
45	1590·43	11·044	55½	2419·22	16·730	74	4300·85	29·936
45¼	1608·16	11·166	56	2463·01	17·104	75	4417·87	30·679
45½	1625·97	11·291	56½	2507·19	17·410	76	4536·47	31·500
45¾	1643·90	11·415	57	2551·76	17·713	77	4656·63	32·335
46	1661·90	11·540	57½	2596·72	18·030	78	4778·37	33·183
46¼	1680·01	11·666	58	2642·08	18·347	79	4901·68	34·039
46½	1698·23	11·793	58½	2687·83	18·665	80	5026·56	34·906

TABLE IV—(*continued*)

Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.	Dia. in In.	Area in Sq. In.	Area in Sq. Ft.
81	5153·00	35·784	87	5944·69	41·282	93	6792·92	47·173
82	5281·02	36·600	88	6082·13	42·237	94	6939·79	48·193
83	5410·62	37·573	89	6221·15	43·202	95	7088·23	49·223
84	5541·78	38·484	90	6361·74	44·178	96	7238·24	50·265
85	5674·51	39·406	91	6503·89	45·165			
86	5808·81	40·339	92	6647·62	46·164			

In calculating the power and speed of governors the following formulae are used :—

Let W — the weight of the revolving body.

,, N — the number of revolutions per minute.

,, R — the radius in feet from centre of motion to the centre of gravity of the revolving body.

Then F, the centrifugal force, will $= \cdot 00033\,WRN^2$.

Having the force, if it is wanted to find the weight required, then—

$$W = \frac{3030\,F}{RN^2} \text{ Weight of Balls.}$$

To find the weight of a cast-iron ball it must first be reduced to cubic inches. This is done by multiplying the cube of its diameter by $\frac{3\cdot 1416}{6} = \cdot 5236$. This gives the number of cubic inches and they must be multiplied by their respective weight, i.e. for cast-iron, ·26 ; wrought-iron, ·28 ; brass, ·3 ; lead, ·41.

The following table gives cubic contents and weights of cast-iron balls.

TABLE V

Diameter in Inches.	Contents in Cubic Inches.	Weight in lb.	Diameter in Inches.	Contents in Cubic Inches.	Weight in lb.	Diameter in Inches.	Contents in Cubic Inches.	Weight in lb.
1	·523	·136	3½	22·45	5·85	8	268·1	69·8
1¼	1·027	·265	4	33·51	8·71	8½	321·5	83·7
1½	1·767	·460	4½	42·71	11·10	9	381·7	99·4
1¾	2·808	·729	5	65·45	17·01	9½	448·9	116·9
2	4·19	1·09	5½	87·07	22·63	10	523·6	136·4
2¼	5·96	1·55	6	113·09	29·40	10½	606·1	157·5
2½	8·18	2·13	6½	143·78	37·5	11	696·9	181·1
2¾	10·89	2·83	7	179·59	46·8	11½	796·3	207·0
3	14·13	3·68	7½	220·90	57·5	12	904·7	235·2

To find the surface of a ball, square the diameter and multiply by 3·1416; or simply multiply the diameter by the circumference—the result is the same.

INDEX

Butler & Tanner, The Selwood Printing Works, Frome, and London.